RAPID EMPLACEMENT AND RETRIEVING DEVICE FOR GROUND STAKES GP-112/G AND GP-113/G

BALLISTIC HAMMER MX6321( )/G

CONTRACT NO. DA-36-039 SC-90760 (E)

U. S. ARMY ELECTRONICS RESEARCH AND DEVELOPMENT LABORATORY
FORT MONMOUTH, NEW JERSEY

AAI

AIRCRAFT ARMAMENTS, Inc.
COCKEYSVILLE, MARYLAND

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SUMMARY REPORT

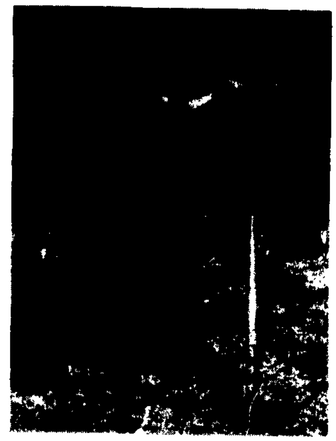
Contract No. DA-36-039 SC-90760(E)

January 15, 1964

Prepared by: R. G. Strickland, Project Manager

C. Gonnerman, Engineer

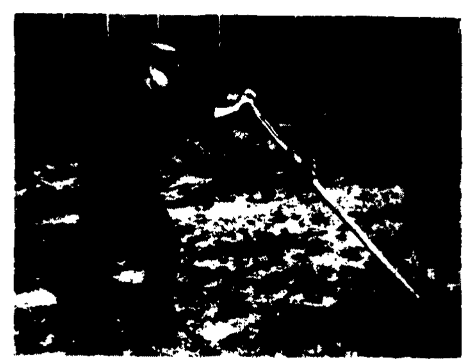
Reviewed by: N. J. LaCosta, Mgr., Explosive Ordnance Dept.

(a) 

 (b)

(c) 

 (d)

 (e)

a. Emplace Stake GP-113/G
b. Retrieve Stake GP-113/G
c. Emplace Stake GP-112/G
d. Retrieve Stake GP-112/G
e. Emplace Stake GP-113/G at approximately 30°

Typical Employment Modes for Ballistic Hammer

Frontispiece

TABLE OF CONTENTS

TABLE OF CONTENTS (cont'd)

I. PURPOSE

The purpose of this program was to conduct a comprehensive background study and development program to establish feasibility of a ground stake emplacement-retrieving device. Capabilities of the device were to include rapid emplacement and retrieve of the Stakes GP-112/G and GP-113/G in soils in which these stakes are normally employed. This was to be achieved with a non-expendable device, minimum weight and volume, and in such a manner as to produce a negligible visual or aural signature.

II. ABSTRACT

Requirements for field mobility and rapid erection of communication masts in the field have made necessary the development of an emplacement-retrieve mechanism which would be fast acting, lightweight, effective in all soils, and be simple to operate and maintain. The fact that the site chosen for erection is dictated by the tactical situation rather than soil conditions makes it necessary that the emplacement-retrieve device be capable of operating where conventional techniques fail.

This program was concerned with the selection of an approach which had the greatest potential to satisfy system requirements and then developing the item to demonstrate the feasibility of the approach selected. A wide range of devices such as explosive shaped charges, rocket devices, recoilless launchers, impact hammers, ballistic tools and vibratory drivers were evaluated and a ballistic hammer selected as having the greatest potential for development.

A design for this device was evolved, based upon analytical considerations and results of tests with a firing mockup. Detail design of this item was described in a Design Plan (AAI Engineering Report 2937) which was submitted to USAELRDL. Following approval of this report, a model was constructed. The performance of this model was evaluated in a series of tests to evaluate driving and retrieving capabilities, effect on ground stake holding power, noise, flash, operator reaction and other pertinent factors. A tool roll to facilitate storage and transportation was also designed and fabricated.

It is believed that the feasibility of the ballistic hammer as a device capable of achieving rapid emplacement and retrieve of ground stakes has been demonstrated by the results obtained during this program.

III. PUBLICATIONS, REPORTS, AND CONFERENCES

During the period covered by the contract, the following publications were prepared:

First Monthly Progress Report,	AAI Engineering Report 2792
Second Monthly Progress Report,	AAI Engineering Report 2792A
First Quarterly Progress Report,	AAI Engineering Report 2843
Phase I Report,	AAI Engineering Report 2853
Third Monthly Progress Report,	AAI Engineering Report 2792B
Fourth Monthly Progress Report,	AAI Engineering Report 2792C
Second Quarterly Report,	AAI Engineering Report 2843A
Design Plan Report,	AAI Engineering Report 2937
Fifth Monthly Progress Report,	AAI Engineering Report 2792D
Sixth Monthly Progress Report,	AAI Engineering Report 2792E
Third Quarterly Progress Report,	AAI Engineering Report 2843B
Seventh Monthly Progress Report,	AAI Engineering Report 2792F
Eighth Monthly Progress Report,	AAI Engineering Report 2792G
Ninth Monthly Progress Report,	AAI Engineering Report 2792H
Tenth Monthly Progress Report,	AAI Engineering Report 2792I
Eleventh Monthly Progress Report,	AAI Engineering Report 2792J
Twelfth Monthly Progress Report,	AAI Engineering Report 2792K
Thirteenth Monthly Progress Report,	AAI Engineering Report 2792L

Conferences between representatives of USAELRDL and AAI were held on a periodic basis to maintain close liaison and as a supplement to the formal written reports, so that project supervision at USAELRDL would be fully informed of progress and direction of work at all times.

Conferences occurred on the following dates:

June 13, 1962 - USAELRDL was visited by R. Strickland and T. Stastny to review general aspects of the stake emplacement retrieving problem with R. Lione and R. Johnson. Mr. R. Lione reviewed the listing of the problem areas and supplied photos of typical installations, stake drawings, and specifications.

July 6, 1962 - USAELRDL was visited by R. Strickland to review initial progress with R. Lione. The plan which AAI proposed to follow under this contract was reviewed and a detailed outline for the feasibility study was presented. Twelve possible emplacement retrieving systems were discussed, as well as evaluation criteria which would be used to determine the most promising systems.

September 20, 21, 1962 - AAI was visited by Mr. R. Lione to review status of work to date. A rough draft of the first quarterly progress report had been prepared and was employed as a basis for discussion. A list of comments by Mr. Lione was compiled and necessary changes incorporated into the quarterly report. AAI was represented in the discussions by R.Strickland, T. Stastny, and N. J. LaCosta.

October 19, 1962 - USAELRDL was visited by R. Strickland to review changes to the first quarterly report and to review the draft of the Phase I report.

December 5, 1962 - AAI was visited by R. Lione and R. Johnson. The design layout for the device was examined and suggestions made concerning various detail considerations. Four test firings with the ballistic test model were demonstrated at the same time.

January 30, 1963 - USAELRDL was visited by R. Strickland and A. Powell to review the design plan and deliver the visualization data. It was agreed at this conference that USAELRDL would accept the plan, but AAI would cover three areas in greater detail and resubmit the plan. The areas expanded upon were: retrieve approach, tool roll carrier, and test schedule.

February 25, 1963 - USAELRDL was visited by R. Strickland, N. J. LaCosta and A. Powell. A presentation was made by R. Strickland which was attended by representatives of the commanding officer, chief scientist, chief engineer and branch chiefs. The presentation included films of tests conducted by AAI and a demonstration of the breadboard model.

March 29, 1963 - USAELRDL was visited by R. Strickland and A. Powell. Detail drawings were reviewed with R. Lione and a film was shown

illustrating an improved retrieve technique.

May 6, 7, 1963 - AAI was visited by R. Lione. The test model was demonstrated in clay and soft rock. Performance was generally satisfactory although difficulty was experienced with the .30 caliber carbine action. R. Lione handled tool during some tests to evaluate operator reaction.

June 18, 1963 - AAI was visited by R. Lione, R. Johnson, P. Devriotes. The device was demonstrated in various soil types. Emplacement was generally satisfactory, but retrieve difficulties were noted.

July 10, 1963 - USAELRDL was visited by R. Strickland, N. J. LaCosta and V. Unger to review new retrieve device design and discuss rescheduling contract dates. R. Lione, R. Johnson, and S. Adler represented USAELRDL.

July 17, 1963 - USAELRDL was visited by R. Strickland, and N. J. LaCosta to review new dates and revised contract scope. R. Lione and R. Johnson represented USAELRDL.

September 11, 12, 13, 1963 - AAI was visited by R. Lione to observe tests with the device. Holding power tests and emplacement and retrieve times were obtained over a wide range of soil types.

September 16, 1963 - USAELRDL was visited by R. Strickland and N. J. LaCosta. Status of work to that date was reviewed with R. Lione.

September 30, 1963 - USAELRDL was visited by R. Strickland and T. Stastny to discuss techniques which might lead to improved stake driving efficiency in future designs. Test data was presented for review and discussion. R. Lione, R. Johnson, S. Adler, and P. Devriotes represented USAELRDL.

IV. FACTUAL DATA

A. Background Investigation

1. Emplacement Techniques

The initial phase of work conducted under this Contract was concerned with the state of the art as it applies to ground stake emplacement and retrieve, and more specifically, with an analysis of emplacement - retrieve techniques which might be capable of satisfying Signal Corps Technical Requirement SCL-4359, 13 November 1961.

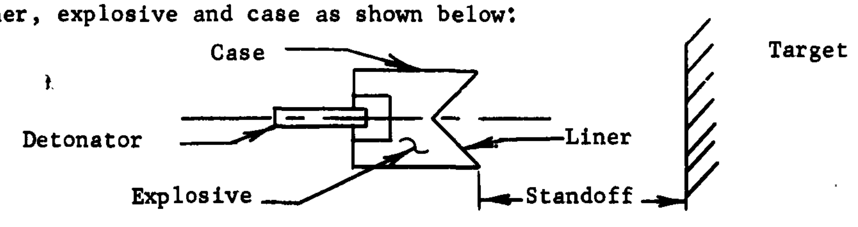
Various methods and techniques for penetrating soil, rock, ice and permafrost were investigated. These were as follows:

Explosive Shaped Charges
Impact Hammers
Rotary Drills
Rotary Impact Drills
Direct Rocket Emplacement
Rocket Torque Wrench
Ballistic Techniques
Vibratory Driving

The paragraphs which follow summarize the characteristics of each technique.

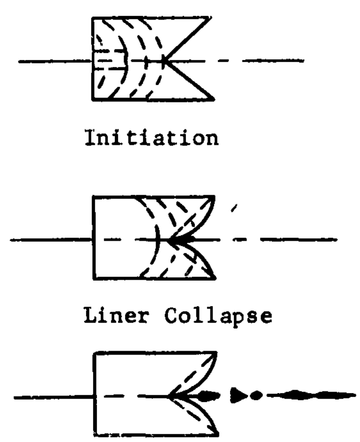
a. Shaped Charges

Shaped charges have been used commercially for such applications as tapping open hearth steel furnaces and tapping pilot holes for explosive mining and blasting. The shaped charge directs the energy of a specially shaped explosive charge to form a high velocity slug or jet capable of extensive penetration. A typical shaped charge consists of liner, explosive and case as shown below:



There has been recent interest in the shaped charge to form holes in earth. The possible employment of this technique to develop holes for ground stakes has been studied.

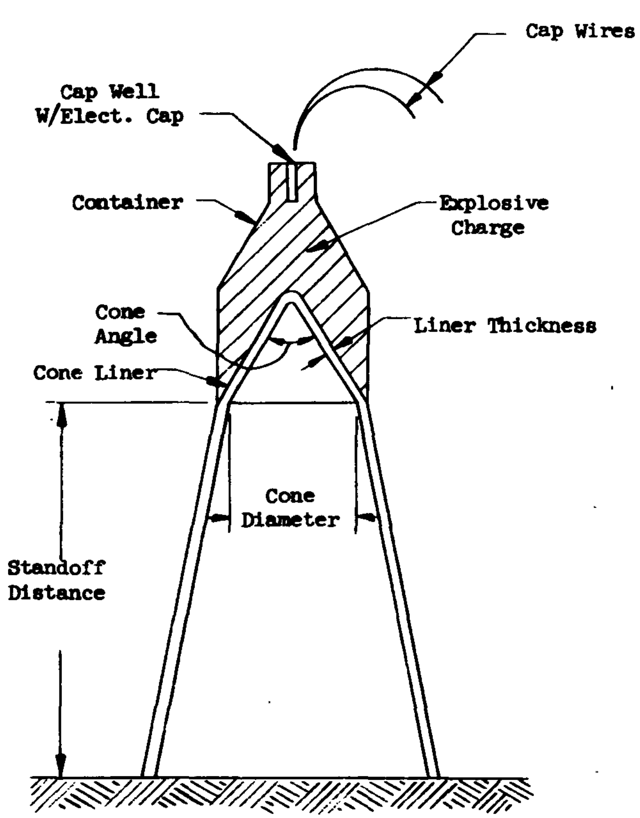
Military applications have been directed to penetration of armor. Extensive testing has been performed to evaluate shaped charge formation of foxholes for the Army in frozen ground (1). Various charges included 5,10,20 and 40 pound charges and a commercial 2-3/4 oz. jet tapper. The best performance of the jet tapper produced a one-inch hole decreasing to 1/2 inch for a depth of 22 inches into frozen ground. A hole ranging from 1-1/2 to 1/4 inch in diameter for a depth of 23 inches was obtained with the jet tapper in fresh water lake ice. The general function of the charge is shown below:



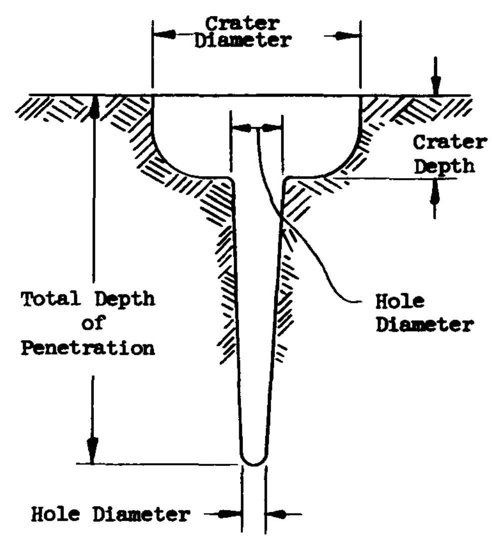
Jet Formation

TYPICAL SHAPED CHARGE FUNCTION

The results are generally consistent, reproducile, and appear desirable for producing holes of these dimensions. Further studies are required in rock formation to establish the extent of fly rock and danger involved to personnel. Figures on the following pages illustrate detail and operational concepts of the shaped charge.



Cross Section of Shaped Charge



Typical Hole Produced by Shaped Charge

Detail Concept of Shaped Charge

Figure 1

AI

PLACING DETONATOR
IN SHAPED CHARGE

F

PREPARING TO EMPLACE STAKE

SH

PLACING DETONATOR
IN SHAPED CHARGE

FIRING SHAPED CHARGE

PREPARING TO EMPLACE STAKE

Figure 2

SHAPED CHARGE EMPLACEMENT CONCEPT

In general, the performance of a shaped charge can be related to the cone angle, cone material, thickness of cone, type of explosive and standoff distance. This is especially true where the strength of the target is insignificant against the jet velocities of around 2000 ft/sec formed by the cone material. Up to a certain point the depth of penetration is theoretically proportional to the length of the jet when the target strength is negligible. Above the optimum standoff distance, the jet tends to break up and penetration is reduced.

The fact that familiarity with blasting techniques is essential and that there is considerable blast, and there may be flying rocks and fragments, greatly decreases the promise of this approach. The associated noise and flash would make it very difficult to prevent enemy observation of the emplacement area.

b. Impact Hammer

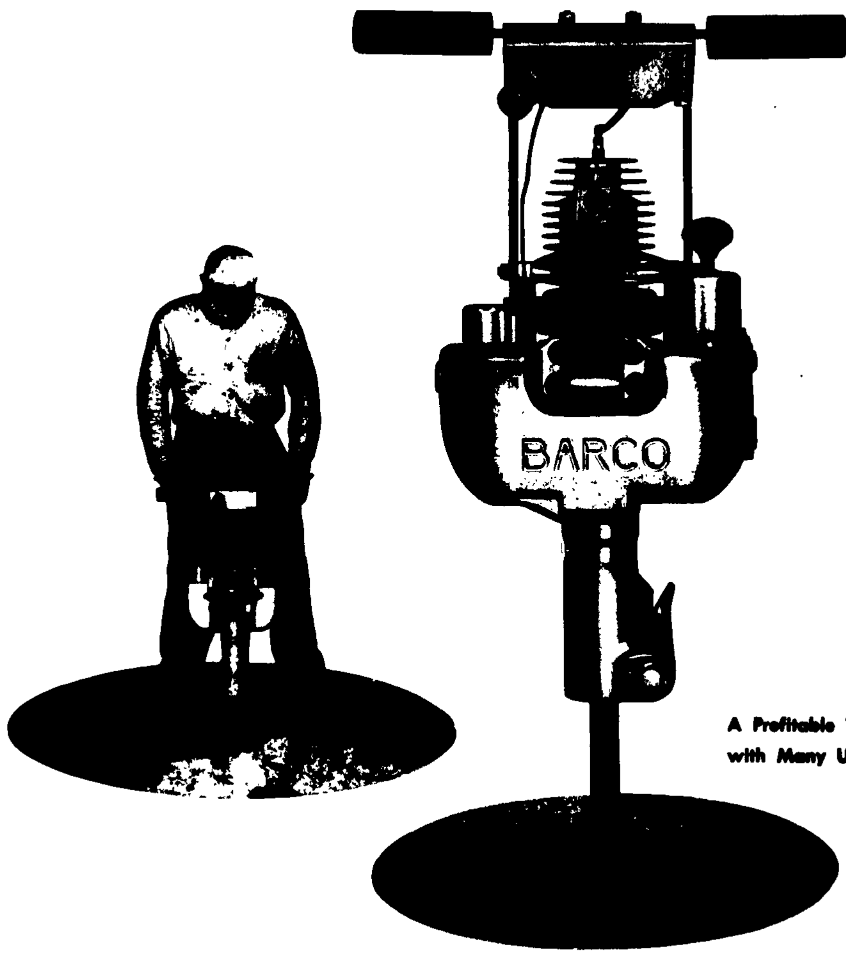
Various commercial impact hammers of one type or another are available, the main difference being the energy source. Perhaps the most familiar is the pneumatic hammer used in breaking concrete. Other impact hammers are gasoline driven, having high impact capability over short strokes, while others are electrically driven. Most are hand held and efficiency is increased by the weight of the operator. Several commercial impact hammers are illustrated on the following pages.

Several tests were conducted by AAI employing an electric impact hammer driving a 1-1/2" star drill into soil conditioned to -65 degrees simulating permafrost. Penetration of 9 inches was achieved in 70 seconds. Increased penetration rate, but a more shallow penetration, was obtained using a wrench to rotate the drill during impact. Using a 1-1/4" star drill a depth of 4 inches was achieved in 15 seconds.

Another type uses blank cartridges to drive various sizes of bolts into wood, plaster, cement, and steel on a one shot basis (2). For small bolt applications this unit is lightweight, compact and highly efficient. Increased performance could well be obtained by automatic or semi-automatic capabilities to deliver more than one impulse to a single

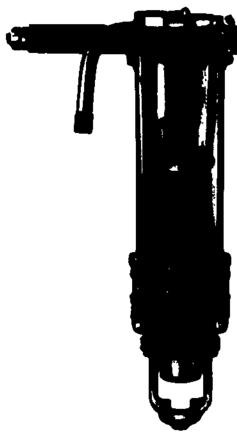
CATALOG 615

BARCO GASOLINE HAMMER M-1



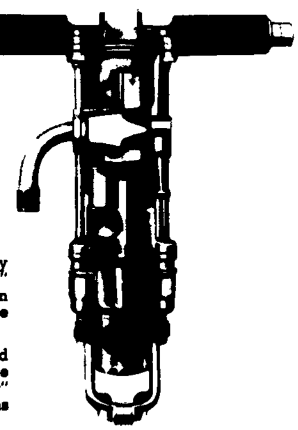
Commercial Gasoline Hammer

Figure 3



This husky mediumweight, a standout in the 45 lb. class, has a conservative capacity of 10 ft. drilling depth . . . or 14 feet under favorable conditions. The L-47 has many construction features in common with the LB-57: Cushion Control . . . rubber-cushioned steel puller . . . full-line-pressure hole blowing . . . built-in oil reservoir . . . replaceable buffer-ring-liner . . . and others. Many parts are interchangeable with the LB-57, which cuts repair parts stocks to a minimum.

With performance approaching the average 55-lb. drill, the L-47 is a universal favorite for general excavating, mining, quarrying and road work. Available in dry or wet types, its field of application includes soft, medium, and medium-hard rock formations.



The L-37 offers "big-drill" performance in the 35-lb. class. This brawny lightweight easily drills to 8 feet and more. It incorporates "big drill" features found in the L-47 and LB-57: one piece steel-puller cushioned on rubber, full-line-pressure hole blowing, built-in oil reservoir, replaceable buffer-ring-liner, and the patented Dual Valve.

The L-37 is a star performer in a wide variety of applications: road building, trench drilling, light ledge drilling, block holing, trimming side walls, cutting hitches, and many more. Best of all, the L-37 is a "miser" with air — an important advantage to the operator when air supply is limited.

Typical Pneumatic Drill

Figure 4

bolt. Speed, simplicity, ability to retrieve stakes, and safety are all associated with the impact device. An operational concept of the AAI Impact (Ballistic)Hammer is found on the following page.

c. Rotary Drills

Rotary drills are available with electric, gasoline, and belt driven energy sources. Many commercial products are on the market for core sampling, well drilling, rock drilling, various soil applications as well as material drilling. These are illustrated on the following pages.

Specific results have been obtained with 1 inch, 1.5 inch, and 2 inch drills in granite under 160 lbs. loading (3). Rotating at a speed of 250 rps the drill is capable of a penetration rate of .3 in./min.

Drill wearing characteristics appear to be strong functions of axial loading. At low loading levels, cutting edges appear to be bouncing over the surface of the rock, resulting in predominatly abrasive action. At high rates of penetration (per revolution of the drill) the bit deteriorates because of overloading of the cutting edges. Chipping of the tool occurs, and the bit suffers an impact failure. Tool chipping phenomenon is apparently evident in hard rocks even at "ideal loadings." Thus, for a given axial load, two performance parameters are evident: rock hardness and rotational speed.

d. Rotary Impact Drilling

The combined effect of rotation and impact drilling has shown increased performance in penetration over a large range of power inputs when compared to either a conventional drill or impact hammer (4). This conclusion was based on experimental evaluation and reported literature survey, where impact penetration is shown to be considerably improved by the mechanical removal of the chips. A rotating flute on the impact drill provides a simple, reliable technique for the removal of any chips tending to reduce the impact effect.

Impulse and frequency of the blow must be correlated with rotational speed and axial load. The bit must withstand imposed loads, and the impact must be such so as not to exceed optimum penetration depths

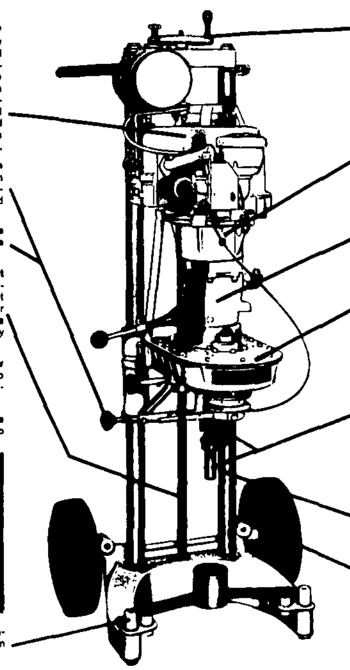
Operational Concept of Ballistic Hammer MX-6321 ( )/G

Figure 5

MINUTEMAN

the first truly PORTABLE, MULTI-PURPOSE ROTARY DRILL

BUILT FOR RELIABILITY AND OPERATING EASE



Lightweight, powerful 5.75 HP Briggs & Stratton aluminum alloy engine is equipped with patented lubrication system and dirt protected, vacuum-type crankcase breather. Heavy-duty construction features include integrally cast cylinder sleeve, heat-treated Armasteel crankshaft, induction hardened crankpin and main bearing journals and aluminum alloy main bearings.

Engine throttle is mounted on power feed lever. Operator can change both engine speed and spindle RPM without removing his hand from lever.

Power feed lever applies engine drive through belt and idler to raise or lower drill head.

MINUTEMAN feed and hoist mechanism is a unique, recirculating ball bearing screw and nut that is 98% friction free. It permits either power or hand feed and prevents jamming even under muddy or dusty operating conditions.

Rolling balls engage both screw and nut to change rotary motion to "up" or "down" motion with minimum friction loss.

Ball raceways in screw and nut are not circular. Grit that works into raceways is forced into spaces formed by diverging arcs where it cannot hinder threading action.

Positive, stake type anchors withstand drilling torque for maximum drill stability and safety.

Hand wheel provides the precise feed control required for diamond core drilling in rock or masonry. Hand wheel also permits continuous 44" stroke in or out of hole whenever manual feed is desirable.

Oversize dry paper type air filter does not cause horsepower loss.

Centrifugal clutch remains disengaged until engine reaches operating RPM, then engages automatically to pick up load smoothly, without grabbing. Unusually simple, durable construction permits clutch to withstand long, hard usage.

Smooth working transmission is an industrial, sliding spur gear type. Gears and shafts are heat treated alloy steel to last the life of the drill.

Rotary housing is a fully enclosed, aluminum alloy casting. It incorporates a pump which supplies a continuous flow of cool oil to transmission and rotary gears regardless of drilling angle.

Clear plastic external circulating oil line lets the operator check oil flow visually.

Rotary housing has two output shafts. The high speed shaft, for rock and masonry coring, has a hollow spindle with "EW" drill rod thread. The low speed shaft is located near drill frame to better withstand heavy loads induced by auger drilling. Auger connections are made by a simple drive pin.

Tough, high tensile steel alloy tube construction insures maximum wearability and lightweight strength.

Large diameter rubber tired wheels are positioned to balance drill weight over axle for easier handling.

Terra-Tire (see photo of core drilling operation on back page) is designed to simplify drill handling on difficult terrain. Furnished as optional item.

SPECIFICATIONS

Dimensions

Height—In.	63½
Width—In.	22
Length—In.	23
Weight—lbs.	235
Domestic shipping, less drilling tools—lbs.	255
Export shipping, less drilling tools—lbs.	335

Optional Engine—Power Products Model AH82 2 cycle engine delivers 6.5 HP @ 5500 R.P.M. Comes equipped with recoil starter, fuel tank and diaphragm carburetor with fuel pump. Recommended for horizontal boring.

Transmission—3 speed forward, 1 reverse

1st gear	3.294-1.0
2nd gear	1.721-1.0
3rd gear	1.00 -1.0
Reverse	3.294-1.0
Lo-Speed Shaft R.P.M. @ 3600 Engine R.P.M.	
1st gear	106
2nd gear	200
3rd Gear	346
Reverse	106
Hi-Speed Shaft R.P.M. @ 3600 Engine R.P.M.	
1st gear	340
2nd gear	650
3rd gear	1110
Reverse	340

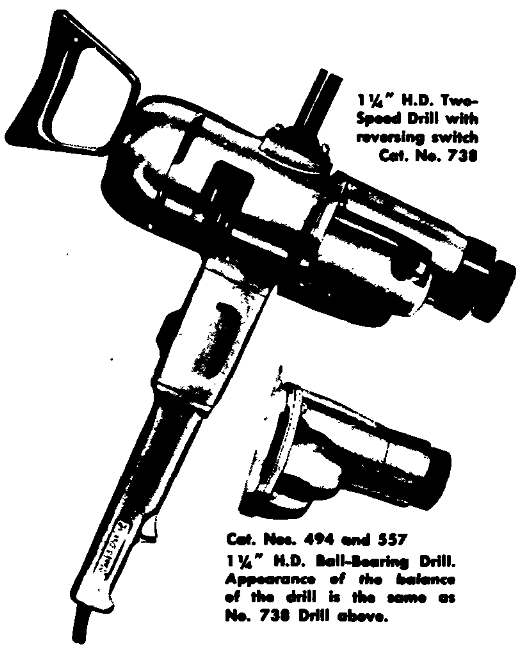
Feed—Recirculating ball bearing screw. Continuous or intermittent power, manual or power in or out, 44" stroke.

Drilling Capacities

Auger Drilling (Lo-Speed Shaft):	
Diameters	3" to 12"
Depth	50' with 3" Augers
Core Drilling (High-Speed Shaft):	
Size	EW
Depth	200'
Masonry Drilling: Bit Diameter	1" to 8"
Horizontal Auger Drilling:	
Diameters	3" to 12"
Depth	50' with 3" Augers

Commercial Soil Drill

Figure 6



1¼" H.D. Two-Speed Drill with reversing switch Cat. No. 738

Cat. Nos. 494 and 557
1¼" H.D. Ball-Bearing Drill. Appearance of the balance of the drill is the same as No. 738 Drill above.

1¼" HEAVY-DUTY BALL-BEARING DRILL

- Top Power, top capacity for drilling or for use as Power Unit.
- Duty Rating—heavy-duty throughout for continuous production service.
- Bearings: 6 ball, 1 lubricant-impregnated.
- Switches — H.D., double-pole, trigger and reversing switch on No. 557.

Cat. No.	Rev. Switch Cat. No.	Cap. with No. 3 M.T.S. Steel	Reaming Capacity Light (Steel)	Reaming Capacity Heavy (Steel)	No-load Speed R.P.M.	Rated Load Speed R.P.M.	Amp. Rating at 115v	Length Inches	Net Wt. lbs.	Ship. Wt. lbs.
494	557	†1¼"	1"	¾"	250	160	12	20½"	32	37½
494-3	557-3*	†1¼"	1"	¾"	350	240	12	20½"	32	37½
494-6*	557-6*	†1¼"	1"	¾"	600	380	12	20½	32	37½
494-9*	557-9*	†1¼"	1"	¾"	900	580	12	20½	32	37½

* Available at extra cost on special order—write for delivery information.

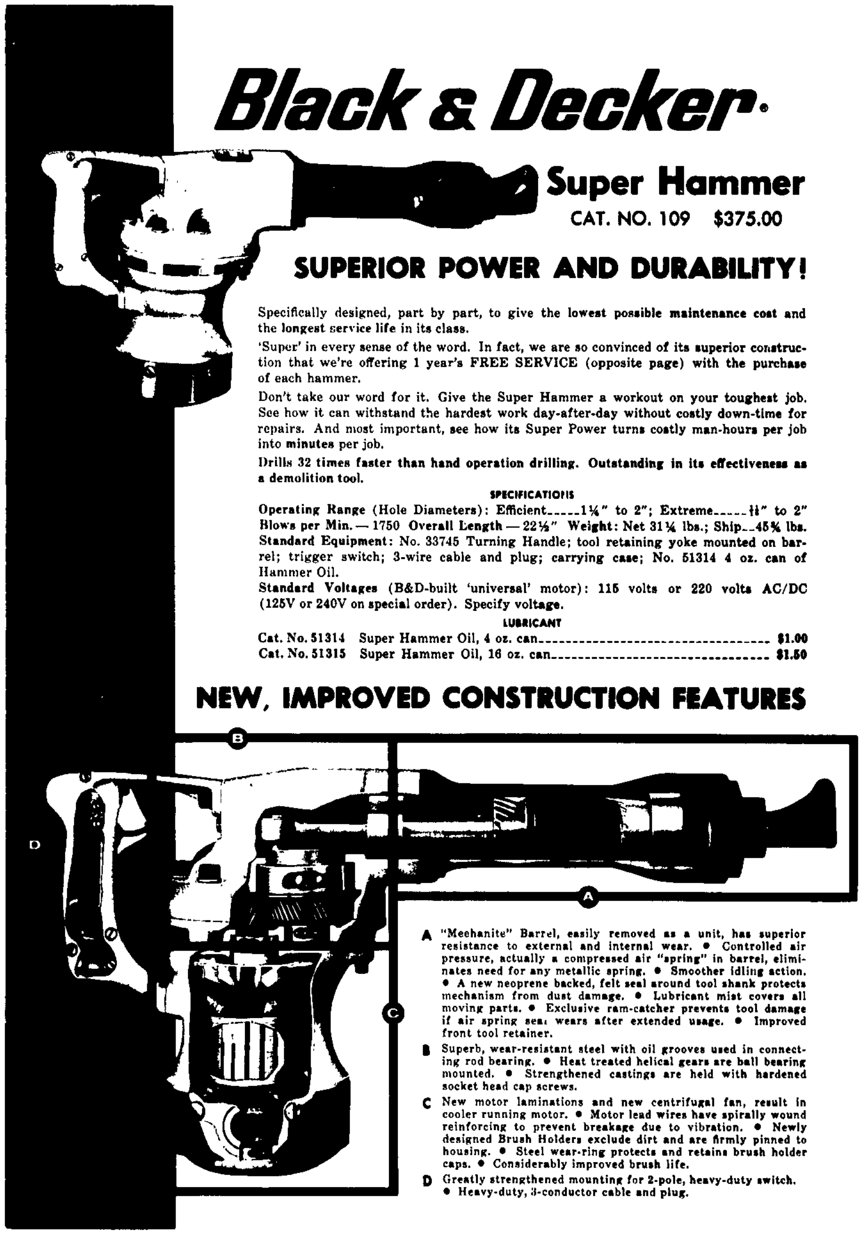
† For drilling small holes: a 3-2 or 3-1 Reducing Sleeve or a Chuck mounted on a Morse Taper Arbor can be used.

Std. Voltages ('universal motor') 115v or 220v AC/DC (others p. 1). Specify voltage.

Std. Equip.: No. 3 Morse Taper Socket; spade handle; detachable pipe handle; 3-wire cable.

Typical Heavy Duty Drill

Figure 7



Electric Percussion Hammer

Figure 8

which would result in rotary stalling. Frequency of the blows, axial loading, and rotational speeds can be optimized for maximum efficiency.

e. Direct Rocket Emplacement

This technique employs a rocket motor mounted on the device to vertically drive the anchor into the ground. While the motor is capable of delivering large thrusts, it is of short duration and difficult to guide during emplacement. Unless coupled with an initial restraint mechanism, it does not have the increased effect from dynamic loading and must produce many times the load of an impact device. An operational concept of this is found on the following page.

Little literature is available on rocket emplacement techniques. One reference employed a modified rocket motor with a small stake and launched it in a tube similar to a bazooka (5). Vertically fired, the rocket motor impacts and detonates a shaped charge in line with the rocket. The shaped charge blasts a pilot hole in advance of the rocket motor reducing the penetration requirements of the motor.

Test results conclude that the device was not suited for use in permafrost and that a special device would be required.

f. Rocket Torque Wrench

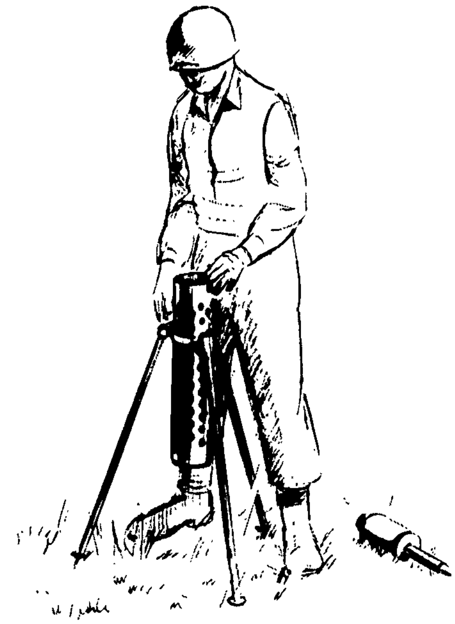
This concept employs a rocket drive to rapidly emplace an auger type anchor stake (6). The major disadvantage is the large forces required to emplace the auger in a short time. By canting the rocket motors a combination of vertical and horizontal thrusts is developed according to the angle of the cant. This is illustrated on the following page.

Emplacement depth is not readily controllable and is strictly a function of burning time and soil encountered. Some guidance device may be required which could release the drive mechanism after a given penetration providing the propellant grain was still burning. This technique could not be employed with the stakes considered under this contract as they require application of axial emplacement force.

g. Ballistic Technique

Penetration studies of projectiles into various mediums

1



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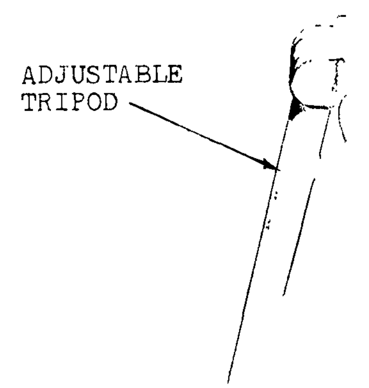

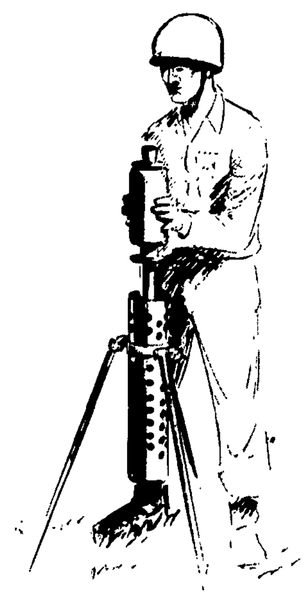

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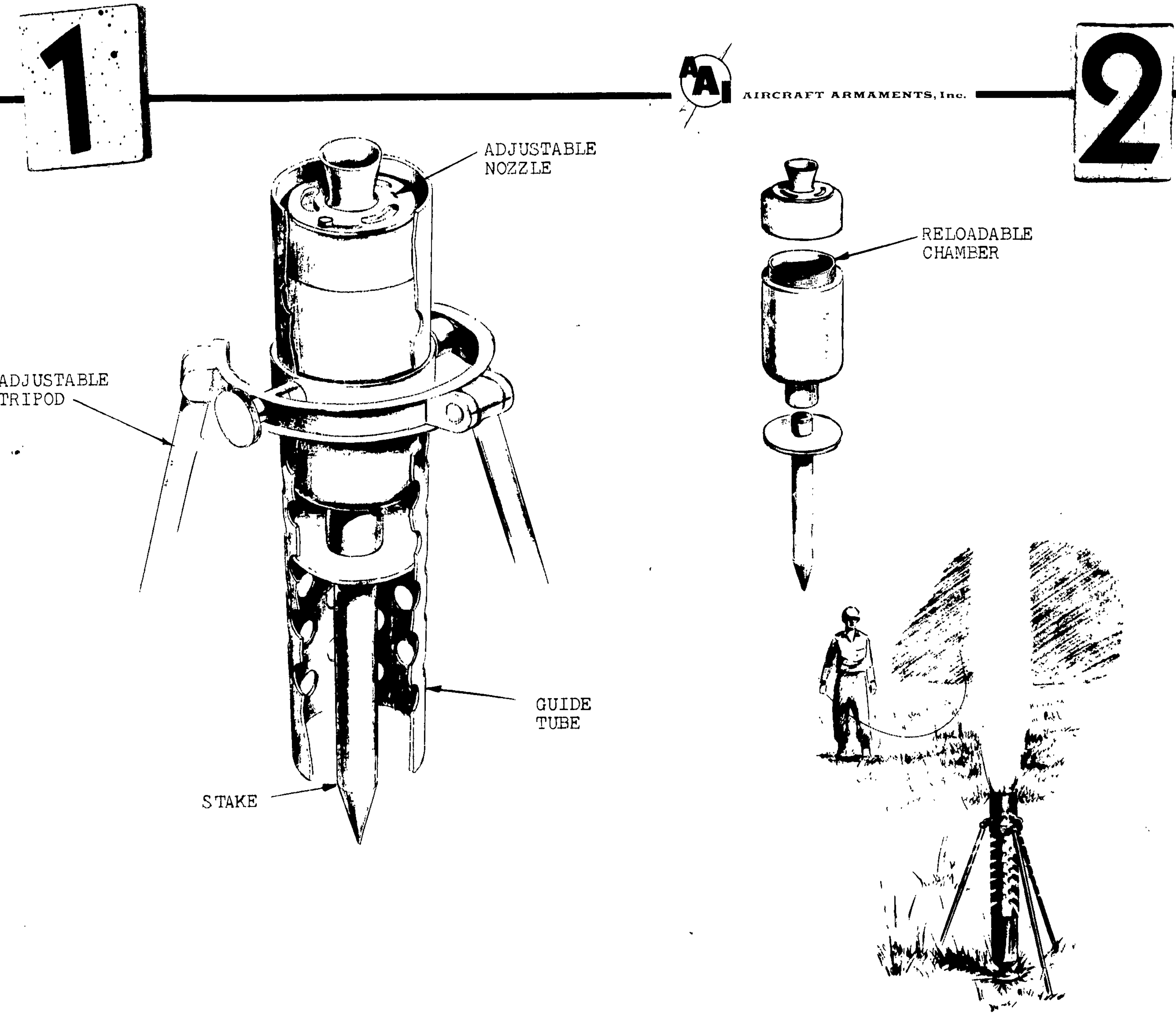
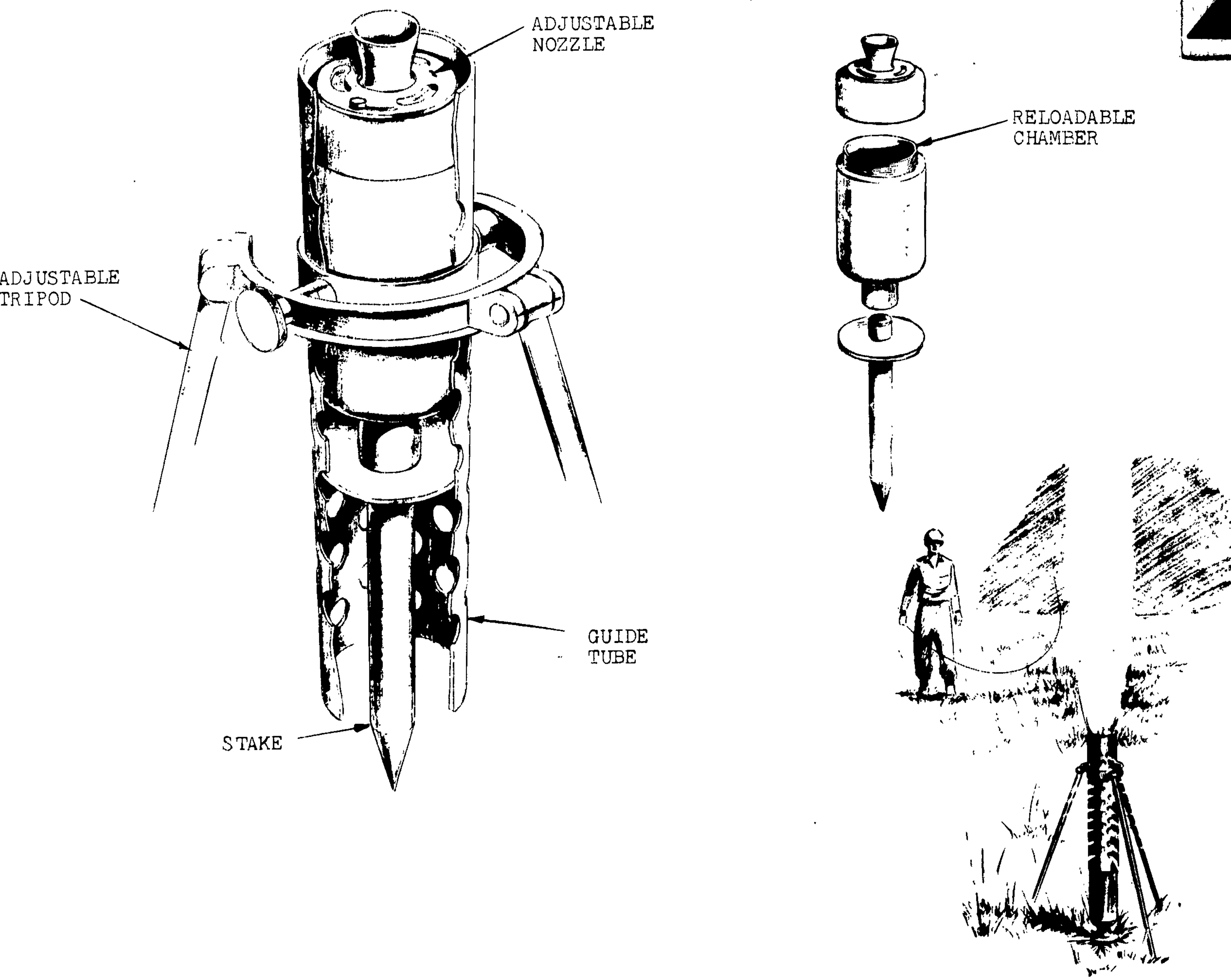


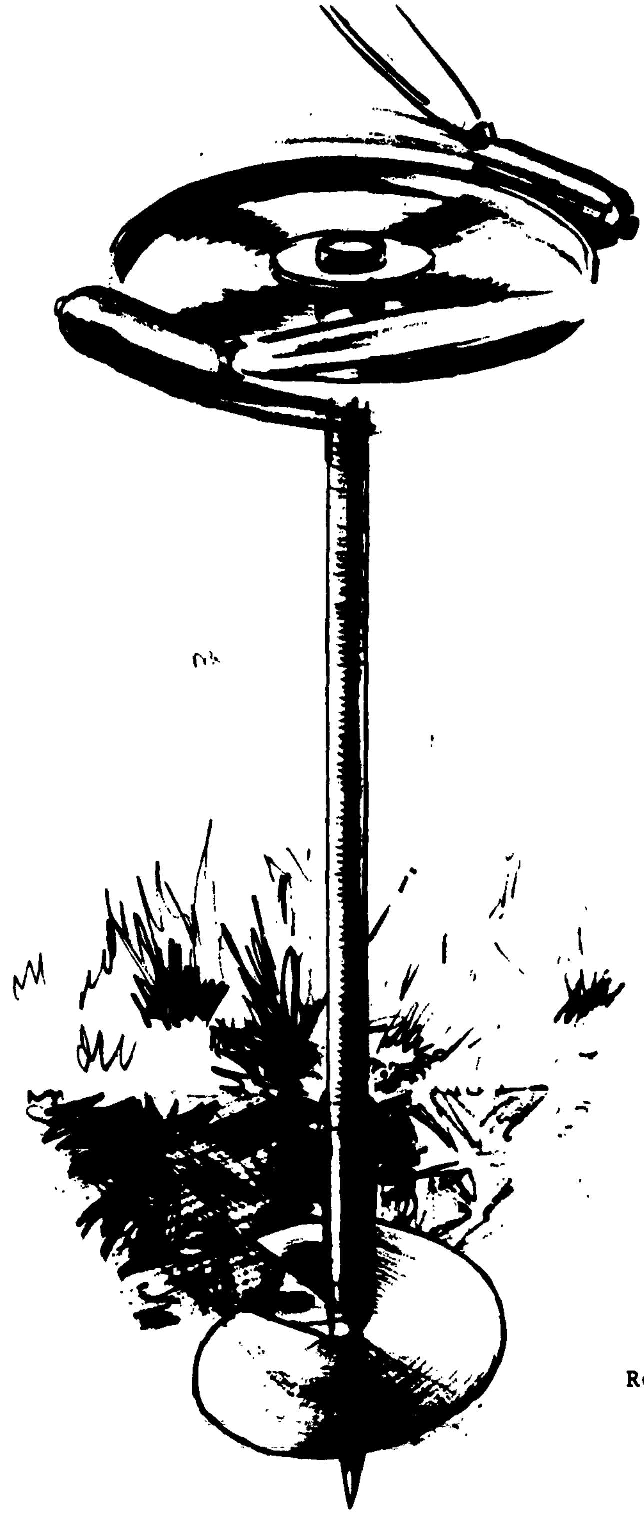
FIRING

Figure 9 ROCKET CONCEPT



FIRING

Figure 9 ROCKET CONCEPT



Rocket Torque Wrench

Figure 10

have been made and reported as early as 1835, known as the famous Metz experiment.

The data resulted in an empirical penetration relation which work at AAI has tended to verify expressed as (7).

$$S = \frac{m}{2bA} \left[\log_e \left(1 + \frac{b}{a} v_o^2 \right) \right]$$

where

S = penetration depth

A = projectile frontal area

m = projectile mass

v_o = projectile impact velocity

a, b = empirical constants depending on soil type

The earliest known penetration equation dates to 1742 where it was assumed that the resistance to penetration was a constant. A more recent paper considers the general equation:

$$\frac{-dv}{dt} = \alpha v^2 + \rho v + \gamma$$

and presents considerable data on sand penetration (14) (15), where α, ρ, and γ are empirical coefficients.

The Ballistic techniques considered were relevant to the analytical investigation discussed above.

The first of these techniques was the silent recoilless. This employs a metal telescoping device which completely retains propellant gas, noise, and flash, yet transmits the propellant kinetic energy during unrolling (under pressure) to a "projectile." This device, which was designed and developed by AAI, is known as the Telecartridge Cup Seal.

The appearance of the Telecartridge Cup Seal is similar to that of a can, the closed end of which has been pushed down through the middle, turning it half-way inside out. Gas pressure generated inside the can will push it back to its original shape. (A barrel tube must be provided

around the outside of the can to prevent bursting under the high pressure generated by the propellant.) The can will completely contain the noise, gas, and flash associated with the propellant ignition. While unrolling, the can will transmit the kinetic energy of the gases to a projectile, pushing it out the barrel. This principle has been successfully applied to several systems of military interest. An operational concept of this principle as applied to stake emplacement is found on the following page.

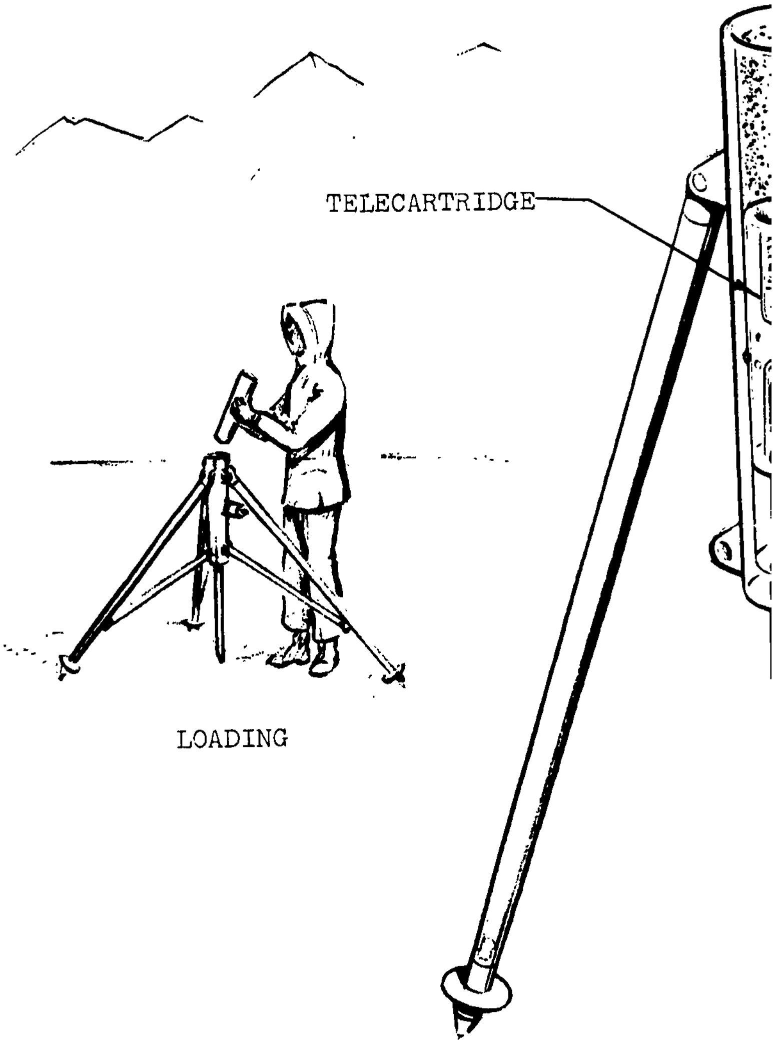
The second of the techniques considered was the recoilless launcher concept. In this concept, recoillessness is obtained by the rearward expulsion of low mass, high velocity gas particles. The design is adapted from the theory of recoilless guns and is straightforward in nature. An operation concept of this technique is found on the following page.

h. Vibratory Emplacement

Increased performance in driving steel piles employing this technique was noted by the Russians and Germans in 1930 (8). More recently this technique was employed in building the bridge over the Yangtze River (9). Extensive experiments showed penetration of various tubular columns as:

	Column Size (dia)	Depths Penetrated	Time, Minutes
In Clay:	5 ft. 1 in.	59 ft.	177
	5 ft. 10 in.	59 ft.	207
	16 ft. 5 in.	54 ft.	180
In Fine Sand:	5 ft. 1 in.	115 ft.	56
	9 ft. 10 in.	75 ft.	14
In Sand & Gravel:	9 ft. 10 in.	53 ft.	113

1



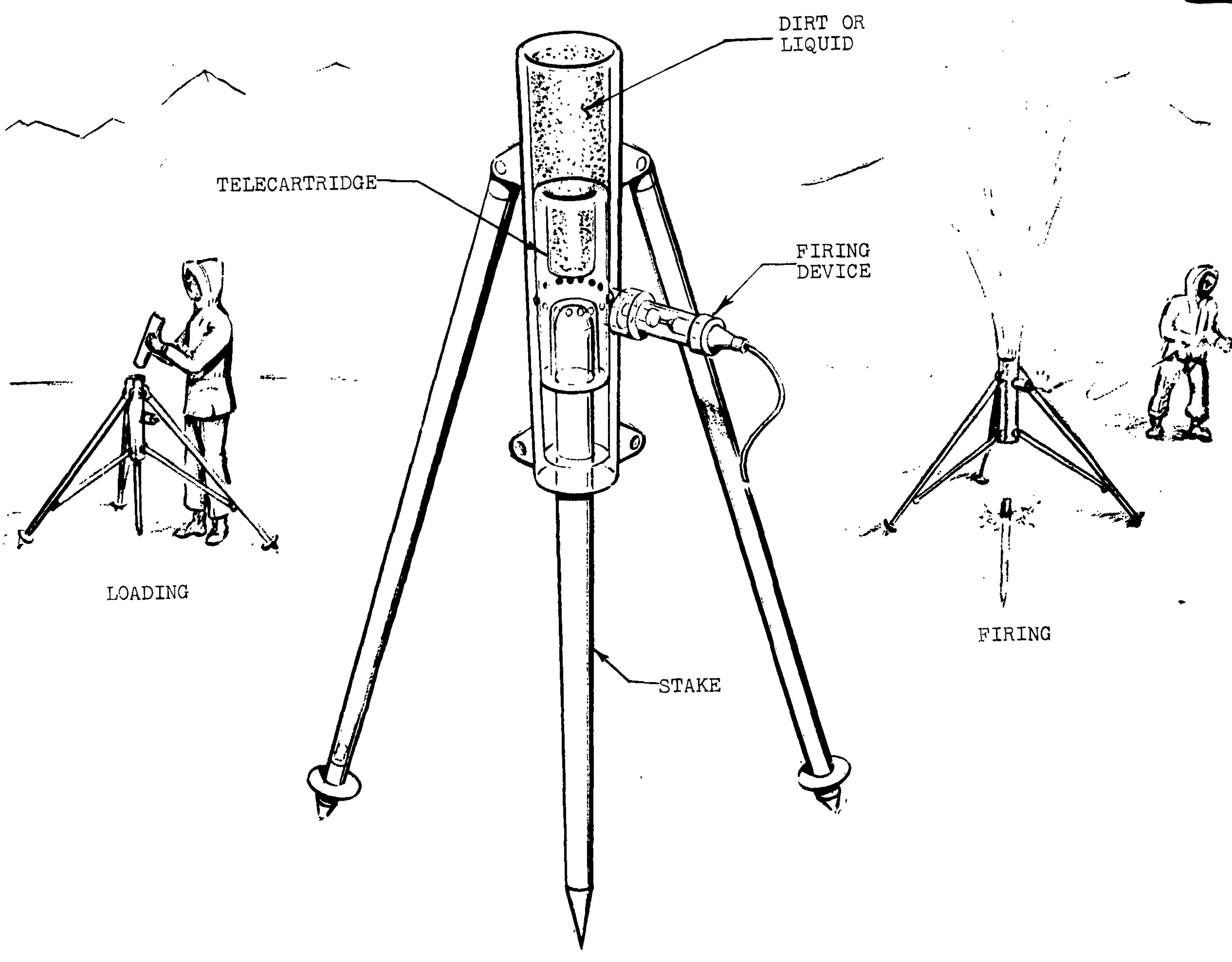
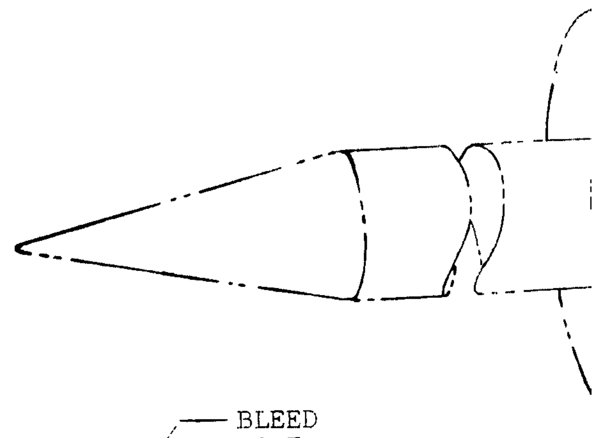
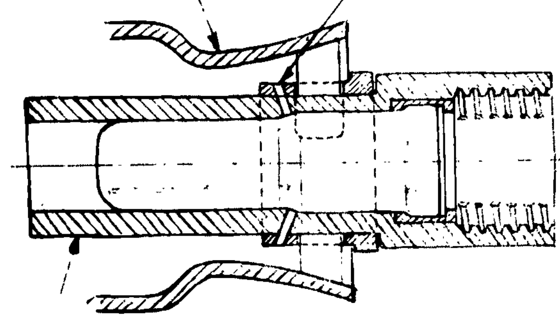


Figure 11

SILENT RECOILLESS CONCEPT

1





DETAIL OF BREECH REGION

POSITIONING

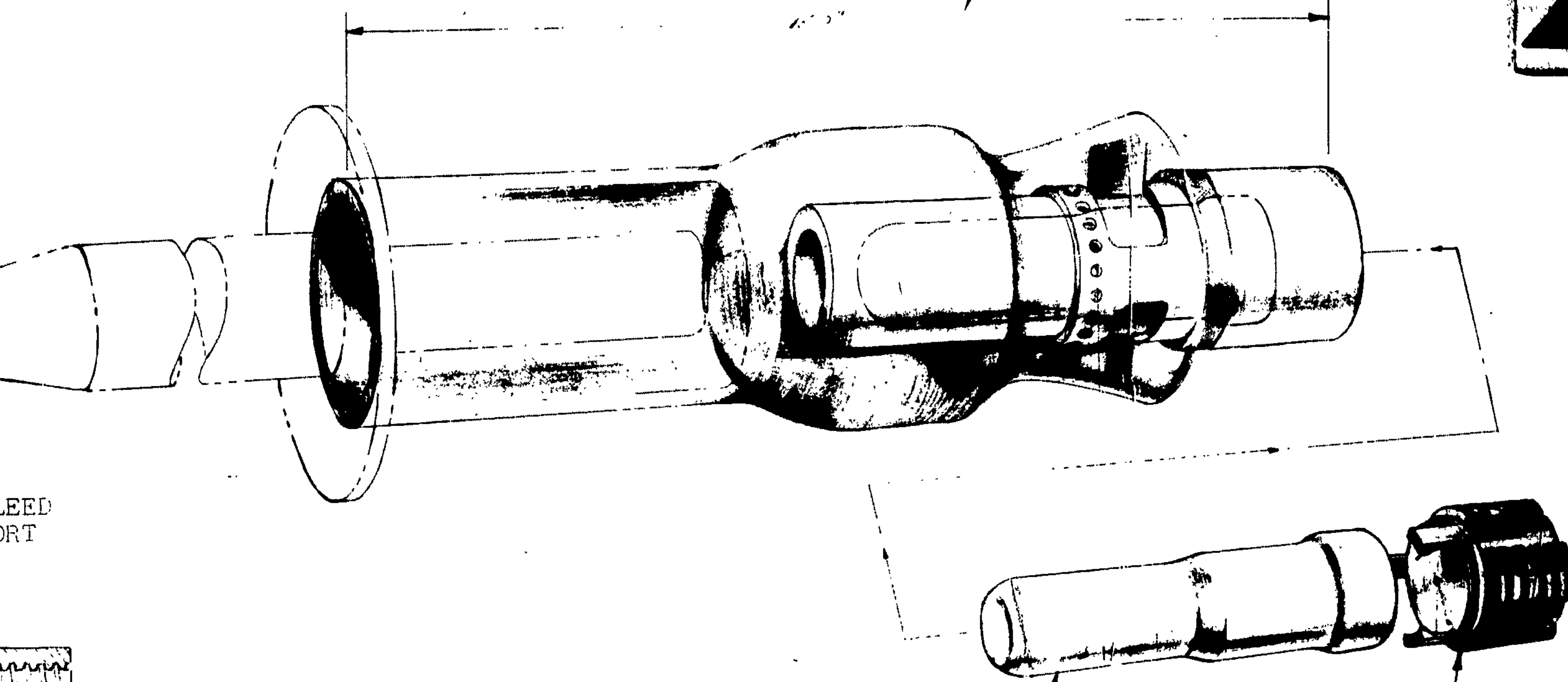
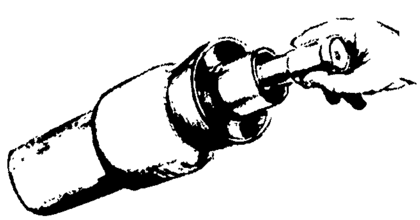
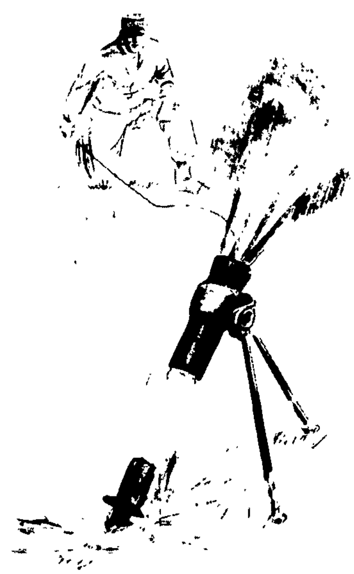


Figure 12
RECOILLESS LAUNCHER
CONCEPT

BREECH



LOADING



FIRING

Currently work is being conducted on the theoretical determination of penetration performance as related to the penetrator geometry, soil properties, vibration forces and frequency, and the system natural frequency (10), (11), (12.)

Successful work in this country has been limited and reported by two commercial firms. The limited success has not been repeatable to a point where a practical lightweight device could be produced with any promise of satisfactory performance.

2. Experimental Evaluation of Commercial Devices

Several commercial devices capable of emplacing ground stakes were tested to gather basic operating data and to evaluate basic emplacement approaches. None of the devices available had a retrieve capability.

Several readily available items were experimentally evaluated and data collected on performance in average soil and simulated permafrost. The items covered included: Gasoline Hammer, Barco Mfg. Co., Model M1; Vibratory-Rotary Electric Drill, Black & Decker No. X-1; Hammer Drill, Black and Decker Model No. 109. It was evident that certain aspects of these devices made them unacceptable for direct application to the stake emplacement retrieve problem. The data obtained was valuable, however, in permitting evaluation of basic approaches. The data obtained in these tests is shown in Figure 13 and 14.

Composition of soil samples employed in all tests was as follows:

Gravel	-	3.5%
Sand	-	46.3%
Silt	-	18.4%
Clay	-	31.8%

Type of Device	Trial No.	Type of Drill Bit	SOIL Type	SOIL Temp °F	POWER Volts	POWER Amps	POWER kw	Penetration (in.)	Time (sec.)	Rate ft/min	Remarks
Black & Decker Rotary-Vibratory	1	2" Earth Auger	Ice	-65	115	7	.805	14	41	1.70	
Experimental Electric Drill Model X-1	2	"	Ice	-65	115	7	.805	14	52	1.35	
	3	"	Ice	-65	115	7	.805	11	47	1.15	
	4	"	Permafrost	-65	115	7	.805	7	196	0.20	Difficulty encountered in extracting Auger.
	5	"	Loam	+80	115	7	.805	26	16	8.15	
	6	1-1/4" masonary drill with Tungsten carbide tip	Permafrost	-65	115	7	.805	5	45	0.55	
Black & Decker Electric Percussion Hammer-Model 109	7	1-1/4" Star Drill Hand oscillation	Permafrost	-65	115	10	1.15	4	14	1.35	Star Drill could not be withdrawn from soil
	8	1-1/2" Star Drill Nonoscillation	Permafrost	-65	115	10	1.15	9	70	0.65	"
	9	1-1/4" Star Drill Hand Oscillation	Ice	-65	115	10	1.15	Nominal	Ice Shattered		

Figure 13 - Tests of Commercial Electric Drills for Preboring Holes

Type of Device	Trial No.	Stake Model	SOIL Type	SOIL Temp. °F	Penetration (In.)	Time (sec.)	Rate ft/min.	Remarks
Barco Mfg. Co. Gasoline	1	GP-112/G	Perma-frost	-65	8-1/2	63	0.67	Penetration stopped after 8-1/2 inches of travel.
Hammer Model No. 1	2	GP-112/G	Perma-frost	-65	12-1/2	136	0.41*	Permafrost cracked on last 2 inches of travel.
	3	GP-112/G	Perma-frost	-65	13	50	9.69*	Permafrost cracked on last 6 inches of travel.
	4	GP-112/G	Ice	-65				No results -ice shattered.
	5	GP-112/G	Perma-frost	-65	10	98	0.51	Penetration stopped after 10 inches of travel.
	6	GP-112/G	Perma-frost	-65	12	123	0.48	
	7	GP-112/G	Perma-frost	-65	13-1/2	128	0.53	

*Adjusted for cracking of permafrost

Figure 14 - Direct Driving of Stakes by Gasoline Hammer

3. Evaluation of Basic Approaches

The approaches and techniques outlined in IV, A, 1 were evaluated to determine the probability of their satisfying the following conditions:

Emplacement into all 8 soil conditions

Emplacement time

Retrieve Time

Weight

Overall Length

Flash and area illumination

Noise

Human Engineering Factors

Safety

Logistics

Maintenance

Simplicity

State of the art (development effort required)

Probable production costs

The items listed were weighted with the items at the top of the list being considered most important.

On the basis of this evaluation, the impact hammer concept scored highest. It appeared at the conclusion of the first phase of work that the impact hammer approach should be feasible and offered the best promise of satisfying the maximum portion of the design requirements.

The silent recoilless concept appeared to offer many desirable features, but was not selected as first choice on the basis of three primary considerations; the approach does not appear well suited to retrieving emplaced stakes; a logistics problem is introduced by the requirement for a recoil mass (sand, dirt, liquid, etc.);and close control of depth of penetration is not achieved.

As a result of weighted evaluation, the following order was established:

Impact Hammer
Silent Recoilless
Vibratory Technique
Sledge Hammer
Hand Auger
Impact Drill
Recoilless Launcher
Rocket Emplacement
Rocket Torque Wrench
Explosive Shaped Charge (*)

B. Soils

In order to be able to attack the general problems of ground stake emplacement and retrieve most effectively, soil classification, soil mechanics and related areas as they applied to stake holding power were investigated.

(*) For complete discussion of methods employed to establish the conclusions, see AAI Engineering Report 2853, Feasibility Study, Contract No. DA-36-039SC-90760.

1. Soil Classification

Soil material is classified in several ways, based upon its penetration characteristics, texture, and the size of its constituent particles. Several classification systems in use at this time are described in the following paragraphs.

The A. B. Chance Co., Centralia, Missouri, has devised a classification system based on the torque required to hand drive an auger into the soil (3).

The torque readings are then correlated to corresponding numerical values ranging from 1 through 8, as follows:

Class	Description of Soil	Probe Value
1	Solid Bed Rock	-
2	Laminated Rock, Slate, Schist	-
3	Shale, Broken Bed Rock, Hardpan	Over 500 in.-lbs.
4	Gravel, Compact Gravel and Sand	400-500 in.-lbs.
5	Medium-Firm Clay, Loose Sand and Gravel, Compact Sand	300-400 in.-lbs.
6*	Soft-Plastic Clay, Loose Coarse Sand, Clayey Silt, Compact Sand	200-300 in.-lbs.
7	Fill, Loose, Fine Sand, Wet Clays, Silt	100-200 in.-lbs.
8	Swamp, Marsh, Saturated Silt, Humus	Under 100 in.-lbs.

*Includes areas only seasonally wet with slow drainage, as in fairly flat terrain.

A similar soil classification adopted from the A.B. Chance description is used by the Signal Corps but differs slightly to include solid ice, coral and frozen ground (13). It is found on the following page.

SOIL CLASSIFICATION CHART

Class	Type	Description
1.	Solid hard rock	
1a.	Solid ice	
2.	Shale, sandstone or soft rock (solid or in layers). Coral, frozen ground	
3.	Hard pan	Hard and dry, requires the use of a digging bar or pick mattock to break the ground.
4.	Crumbling, damp	Consists principally of clay, in a state that will crumble when an attempt is made to squeeze it into a ball with the hand.
5.	Firm, moist	In many cases, clay is predominent although the soil may contain small stones, gravel, or sand and when squeezed with the hand, it will form a firm ball. Most soils in well drained areas, other than hillsides, fall into this classification.
6.	Plastic, wet	In most cases, this soil is predominantly clay as in Class 5. Because of unfavorable moisture conditions, such as areas subjected to heavy seasonal rainfall, sufficient water is present to penetrate the soil to a considerable depth, even though the area may be fairly well drained. During such seasons the soil becomes "plastic" and, when squeezed in the hand will readily assume any shape into which it is molded. This soil is frequently found in flat terrains, or near rivers and marshes.
7.	Loose, dry	Usually found in arid regions where sand or gravel predominate. Filled-in or built-up areas in dry regions, or during very dry seasons, fall into this classification. Lack of bonding material to hold the particles together causes the soil to remain very loose.
	Loose, wet	Same as loose, dry for holding power, although it contains considerable sand, gravel, loam, silt, or very wet clay. In general, this soil resembles mud, and free water will usually drain into an anchor hole while it is being dug. Holding power in favorable seasons may approach Class 5 but, due to the porosity of

SOIL CLASSIFICATION CHART (cont.)

Class	Type	Description
7	Loose, wet (cont.)	the soil, it absorbs excessive moisture during rainy seasons, or high tides, with a resultant loss of holding power. This soil is found extensively in poorly drained areas, and in areas where there is no drainage, although the subsoil may be entirely of clay. This class also includes very soft, wet clay.
8	Swamps, marshes & deep snow	

Information furnished by the Engineering Department of A. B. Chance indicates that the probe cannot be used in below-freezing temperatures where the soil is frozen for a depth of greater than two inches.

It was also indicated that the soil classification method described above will be amended to correlate ground probe readings with the ASTM method of determining the penetrability of soils. The latter utilizes the Swiss Hammer techniques which records the number of blows required to drive a sleeve into a prebored hole.

Another classification method is the "California Bearing Ratio" (14). About 8 pounds of soil sample at optimum moisture content and expected field density are compacted in a cylindrical mold 6 inches in diameter and 7 inches high. The sample is then tested for penetration of a 3-inch square piston in a testing machine. The unit loads required to produce certain penetrations are divided by some standard unit loads established for the same values of penetration. The smallest of several ratios (in percent) thus obtained is the "California Bearing Ratio."

Soil is also classified by the size of its constituent particles and their grading as follows:

Type	Diameter of Particles (MM)
Fine gravel (grit)	2 to 1
Coarse Sand	1 to 0.5
Medium Sand	0.5 to 0.25
Fine Sand	0.25 to 0.10
Very Fine Sand	0.10 to 0.05
Silt	0.05 to 0.005
Clay	0.005 and below

Sands and gravel are non-plastic in the wet state and possess no strength or cohesion between individual mineral particles or rock fragments in the dry state.

As the particle size decreases the soil becomes more cohesive. Clay, for example, is "moldable" or plastic and sometimes sticky in the wet state. It possesses sufficient strength in the dry state to form hard lumps which cannot be pulverized by the fingers and is difficult to indent with the thumbnail.

Soils are also classified on the basis of combinations of the above types with or without the presence of water and organic material.

Mud: A slimy or pasty mixture of soil and organic admixtures at the bottom of a river or lake.

Gumbo: A fine-grained soil usually devoid of sand, impervious when saturated, and waxy to the touch.

Adobe: A heavy-textured alluvial clay found mainly in the semi-arid regions of the Southwestern United States.

Marl: A clay with calcium carbonate.

Hard pan: A lightly compressed and cemented mixture of sand, clay, gravel and boulders, generally located on top of ledge rock. Excavation by pick is difficult.

Peat: Partly decayed plant material. Impure peat, containing considerable inorganic admixture is known as "muck", while organic deposits in small vestigial lakes are peat bogs or muskegs.

Loam: A mixture of sand, silt and clay.

Permafrost

In a category by itself relative to the emplacement of anchoring devices is permanently frozen soil or "permafrost," as it is more popularly known (15).

It merits separate consideration not only for the problems involved in the physical penetration of the soil, but because of the stringent conditions imposed on personnel and equipment due to ice, snow, and low atmospheric temperatures.

Human engineering considerations must recognize that operating personnel will be required to wear heavy, bulky clothing, including gloves or mittens which should influence the design of emplacement tools and equipment.

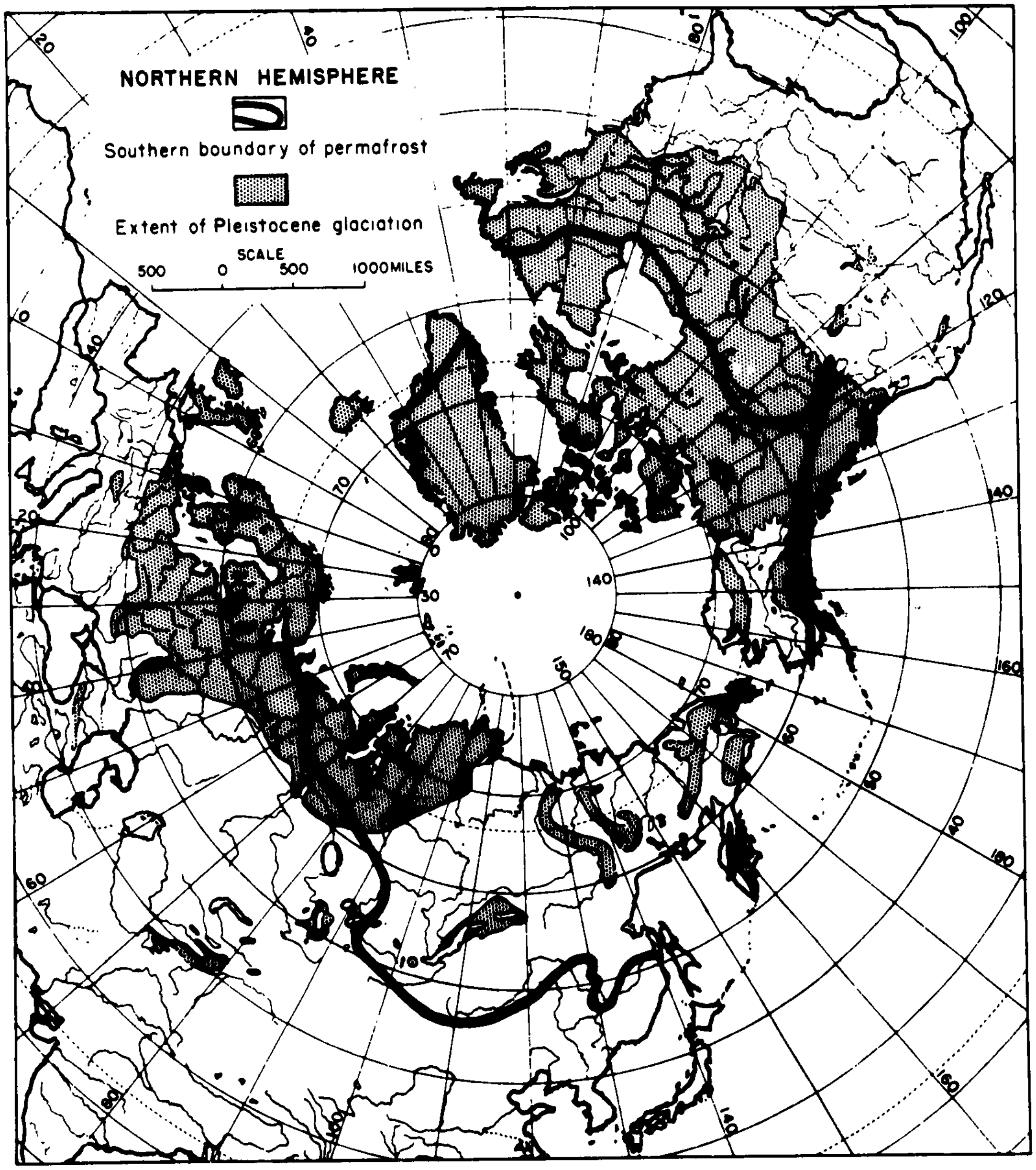
Permafrost is defined as a thickness of soil or other superficial deposits or even bed rock, at variable depths beneath the surface of the earth. Here temperatures below freezing have existed for long periods of time (from two to thousands of years). Permanently frozen ground is defined exclusively on the basis of temperature, irrespective of texture, degree of induration, or water content.

A dry frozen condition is usually found in sandy or in other coarse-grained material that drains easily. Dry frozen ground has neither the hardness nor the induration which are characteristic of permanently frozen ground containing moisture in the form of ice.

Permafrost underlies about one-fifth of the entire land surface of the world. It is most widespread in the northern hemisphere around the shore of the Arctic Ocean (See Figure15) but is also extensive in the Antarctic.

The physical properties of frozen ground depend on its composition, texture, content of ice and temperature. There is yet no classification in which all of these factors are taken into account.

Ice fills some or all of the interstitial space between the mineral grains in frozen ground and acts as cement. Mechanical properties of frozen ground, therefore, tend to approach those of ice.



Permafrost Distributing, Northern Hemisphere

Figure 15

The relative strength of ice under different stresses depends on its structure, temperature and the nature of surrounding conditions.

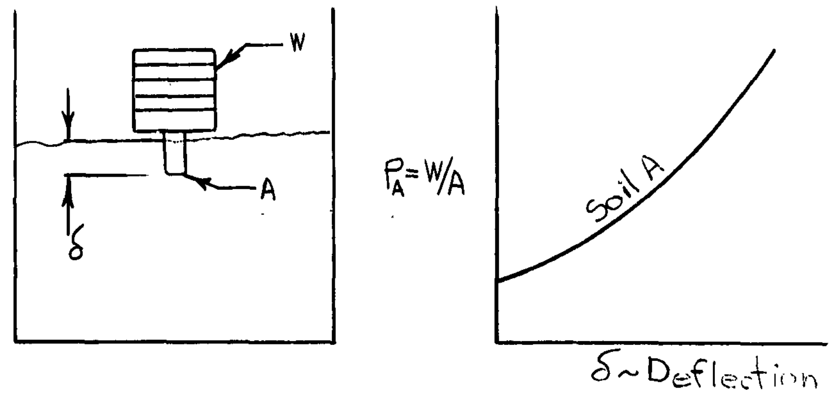
In North America, the boundary of the permafrost areas roughly follows the course of the Yukon River, then runs easterly along the 60th parallel to the 110th meridian, where it swings southwest toward the south end of Hudson Bay. From the Hudson Bay the boundary line turns northeastward and reaches the Atlantic Ocean on the east coast of Greenland, opposite Iceland.

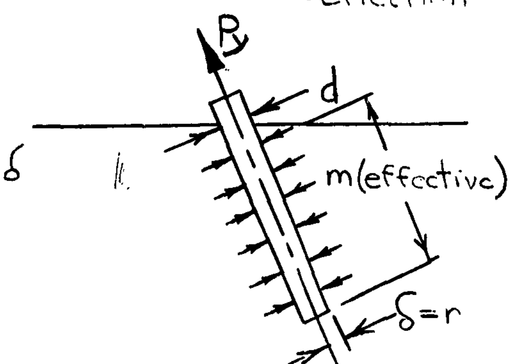
The thickness or depth of penetration of permafrost is usually several yards, but in many places, especially along the coast of the Arctic Ocean, it exceeds one or two hundred yards.

2. Holding Ability, Ground Stake

A theory and equations are presented describing the holding power of a stake anchor as a function of the permissible creep and the earth resistance at the deflection. Two separate expressions are given; one describing the maximum force that can be applied axially to withdraw the stake, and one applied perpendicularly to deflect the stake. The stake is assumed to be infinite in strength and rigidity and changes in load orientation are neglected with changes in creep. Further, creep is limited to where the soil is restrained from flowing around the periphery; on the order of 1/4 - 1/3 the diameter.

The theory is simplified with empirical coefficients providing a useable rapid technique for estimating the holding power of a stake with data obtained from a pressure deflection curve in an unrestrained soil sample. (See sketch on following page).





Maximum Axial Load

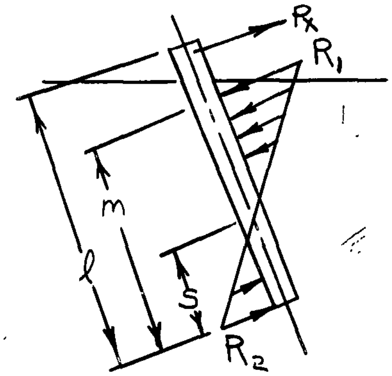
Driving a stake into the soil causes the soil to deflect a distance δ radially, equal to the radius of the rod. This deflection gives rise to a compressive stress encompassing the submerged portion. Depending on the lacerations, spiral or finish on the rod the force P_y will be given as:

$$P_y = \psi \, m \left(\pi d \right) P'_g \qquad (1)$$

where

P_y	=	force required to extrace stake	lbs.
ψ	=	surface finish coefficient	-
m	=	depth of emplacement	in.
d	=	stake diameter (2r)	in.
P'_g	=	soil shearing stress or uniform ground pressure at δ = r	lb./in.^{2}
δ	=	ground deflection	

MAXIMUM TRANSVERSE LOAD



When a transverse load P_x is applied to the stake, a rotation about the point at s is assumed causing linear deflections in the soil. Each linear deflection is resisted by a force proportional to the deflection resulting in a linear load distribution of R lb/in. where R = $P_G d$. Here P_G is measured at δ = r + δc and δc is the permissible creep.

Summing forces and moments:

$$\Sigma F_x = 0 = P_x - (1/2)R_1 (m-s) + (1/2)R_2 s \qquad (2)$$

$$\Sigma M_2 = 0 = P_x \ell - (1/2)R_1(m-s)\left[\frac{2}{3}(m-s)+s\right]+(1/6)R_2 s^2 \qquad (3)$$

$$\Sigma M_1 = 0 = P_x (\ell - m)+(1/6)R_1(m-s)^2-(1/2)R_2 s(m-1/3s) \qquad (4)$$

From the geometry:

$$s = R_2 m/R_1 + R_2 \qquad (5)$$

Solving equation (2) for s,

$$s = \frac{R_1 m - 2 P_x}{R_1 + R_2} \qquad (2.1)$$

Equating (2.1) with (5) for R_2

$$R_2 = R_1 - \frac{2 P_x}{m} \qquad (6)$$

Solving equation (3) for S

$$0 = 6P_x \ell - R_1 \left[2(m^2 - 2ms + s^2) + 3(ms - s^2)\right] + R_2 s^2$$

$$0 = 6P_x \ell - R_1 \left[-s^2 - ms + 2m^2\right] + R_2 s^2$$

$$0 = (R_1 + R_2) s^2 + R_1 ms - 2R_1 m^2 + 6P_x \ell \qquad (3.1)$$

Substituting equation (5) in (3.1) and solving for R_2

$$0 = \frac{R_2^2 m^2}{R_1 + R_2} + \frac{R_1 R_2 m^2}{R_1 + R_2} + 6P_x \ell - 2R_1 m^2$$

$$0 = R_2^2 m^2 + R_1 R_2 m^2 + (6P_x \ell - 2R_1 m^2) R_1 + (6P_x \ell - 2R_1 m^2) R_2$$

$$0 = R_2^2 + \left(\frac{6P_x \ell}{m^2} - R_1\right) R_2 + \left(\frac{6P_x \ell}{m^2} - 2R_1\right) R_1 \qquad (7)$$

Substituting equation (6) in (7) to solve for P_x

$$0 = R_1^2 - \frac{4P_x R_1}{m} + \frac{4P_x^2}{m^2} + \frac{6R_1 \ell}{m^2} P_x - \frac{12P_x^2 \ell}{m^3} - R_1^2 + \frac{2R_1 P_x}{m}$$

$$+ \frac{6P_x R_1 \ell}{m^2} - 2 R_1^2$$

$$0 = 4\left[\frac{P_x}{m}\right]^2 \left(1 - \frac{3\ell}{m}\right) + \frac{P_x}{m} \left(12 R_1 \frac{\ell}{m} - 2R_1\right) - 2R_1^2$$

$$= \left(\frac{3\ell}{m} - 1\right) \left[\frac{P_x}{m}\right]^2 - \left[3 R_1 \frac{\ell}{m} - \frac{R_1}{2}\right] \frac{P_x}{m} + \frac{R_1^2}{2}$$

$$\frac{P_x}{m} = \frac{\frac{(6\ell - m)R_1}{2m} - \sqrt{\frac{(36\ell^2 - 12\ell m + m^2) R_1^2}{4m^2} - \frac{(6\ell - 2m)}{m} R_1^2}}{\frac{2}{m}(3\ell - m)}$$

$$= \frac{(6\ell - m)\frac{R_1}{2m} - \frac{R_1}{m}\sqrt{9\ell^2 - 9\ell m + \frac{9}{4} m^2}}{\frac{2}{m}(3\ell - m)}$$

$$= \frac{R_1\left[\frac{(6\ell - m)}{2} - (3\ell - \frac{3}{2}m)\right]}{2(3\ell - m)}$$

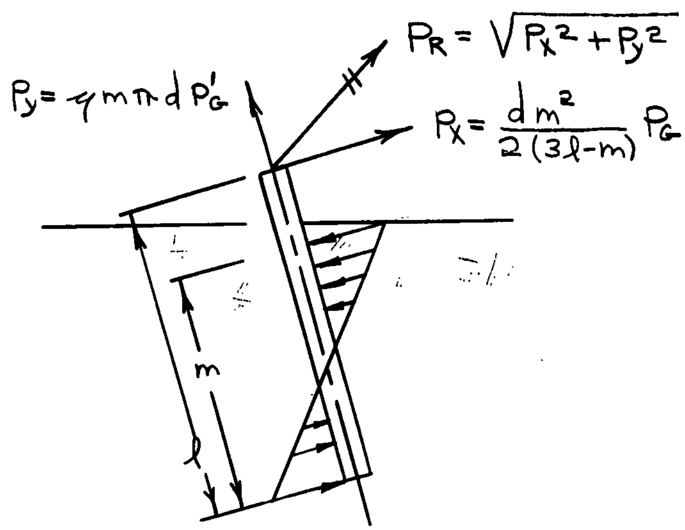
$$\frac{P_x}{m} = \frac{R_1 m}{2(3\ell - m)}$$

$$\therefore P_x = \frac{R_1 m^2}{2(3\ell - m)}$$

$$\therefore P_x = \frac{P_G d m^2}{2(3\ell - m)}$$ P_G at deflection permissible at R_1

Therefore, the maximum holding strength of the stake is the resultant of P_x and P_y as shown on the following page.

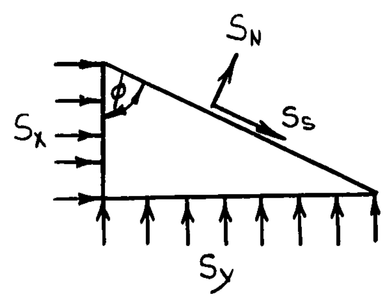
Though the equations are presented in a readily applicable form, the problem is not reduced to a handbook type solution. Good engineering practice and experience must be applied in the design of a ground stake to account for wide variations from the assumed soil properties. While soil properties are not expected to vary over the depth of these



shallow stakes, they are expected to change over prolonged period of seasonal changes and settling or strength reduction under repeated loadings. Likewise, imperfections in the homogeniety of the medium must be considered.

Where little deflection occurs before rupture in such brittle materials as rock and ice, the pressure P_G may well be the maximum compressive stress of the material. If instead of local failure under compression, the material shears, the stress P_G can be computed from:

$$(-S_s) = \frac{S_x - S_y}{2} \sin 2\phi$$



where;

S_x, S_y : normal stress

S_s : max. shear stress

ϕ : plane of shear failure

Here

$S_y = 0, \quad S_x = P_G$

C. Detail Design

1. Emplacement Device

a. Basic principles of operation

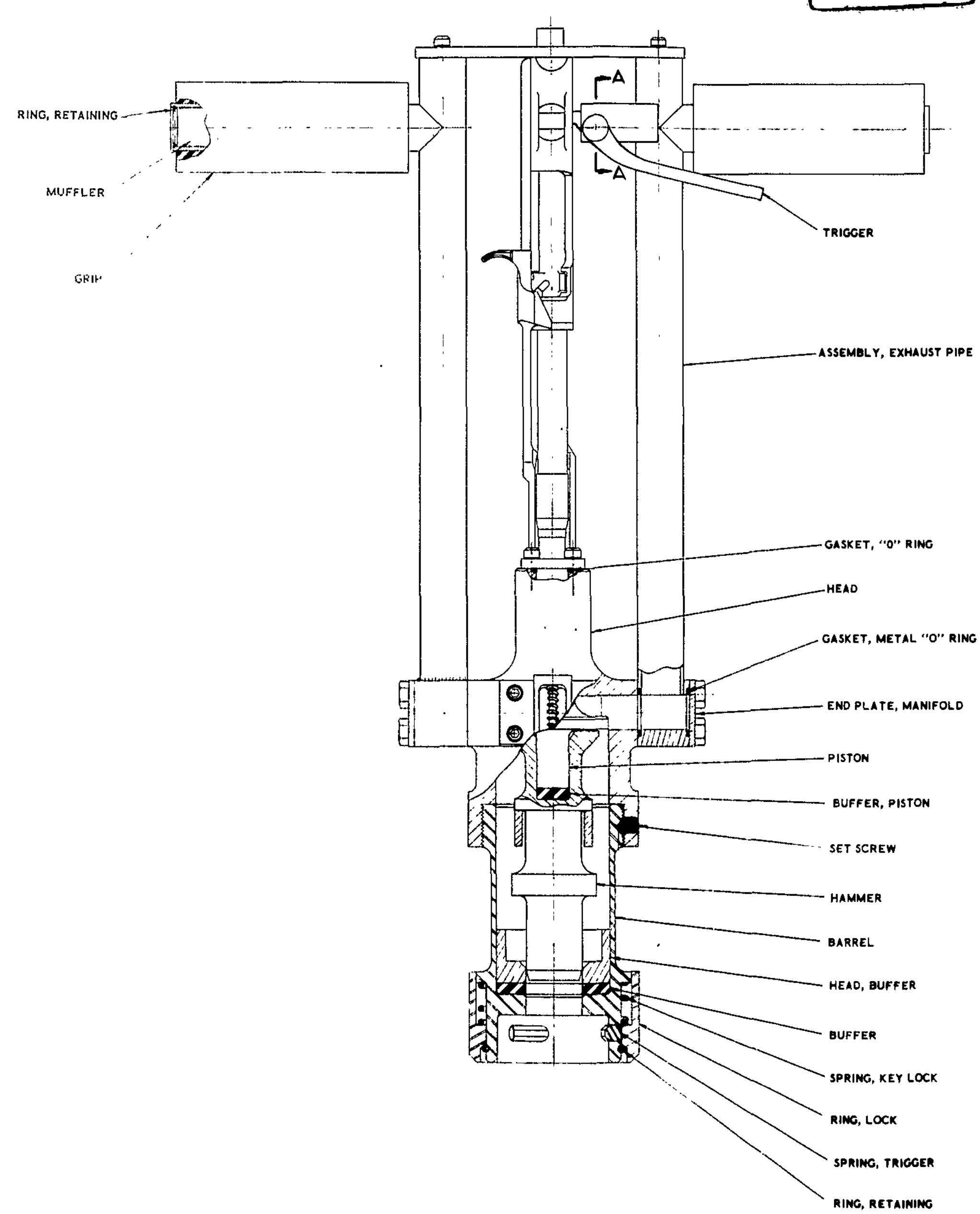
As has been described in Section IV. A. of this report, many techniques may be considered to emplace or retrieve ground stakes. The study of the feasibility of these approaches indicated that the most effective technique would be one which employed repeated blows of a piston. In most respects the Ballistic Hammer is similar in principle to the sledge hammer; a weight is accelerated to a velocity to impart energy to a stake as the result of impact between weight and stake head. The means employed to propel the weight to the required energy varies considerably in the two techniques. In the sledge hammer, human muscles are employed to lift the weight above the head of the operator and to accelerate the hammer head through a downward "work" stroke. In the Device, however, the high pressure gas, resulting from the ignition of a propellant charge, is used to impart energy to the weight. It utilizes multiple blows since a single dynamic load would transmit an intolerably large recoil impulse to the operator and a greater measure of control is introduced by discrete energy pulses rather than a single "one shot" pulse.

b. Description and functions of Components

While the basic operating principle is extremely simple, the incorporation of this into the Device results in a unit of reasonable complexity. A layout drawing is shown in Figure 16. The Device incorporates the following features to provide the effects of multiple dynamic loadings:

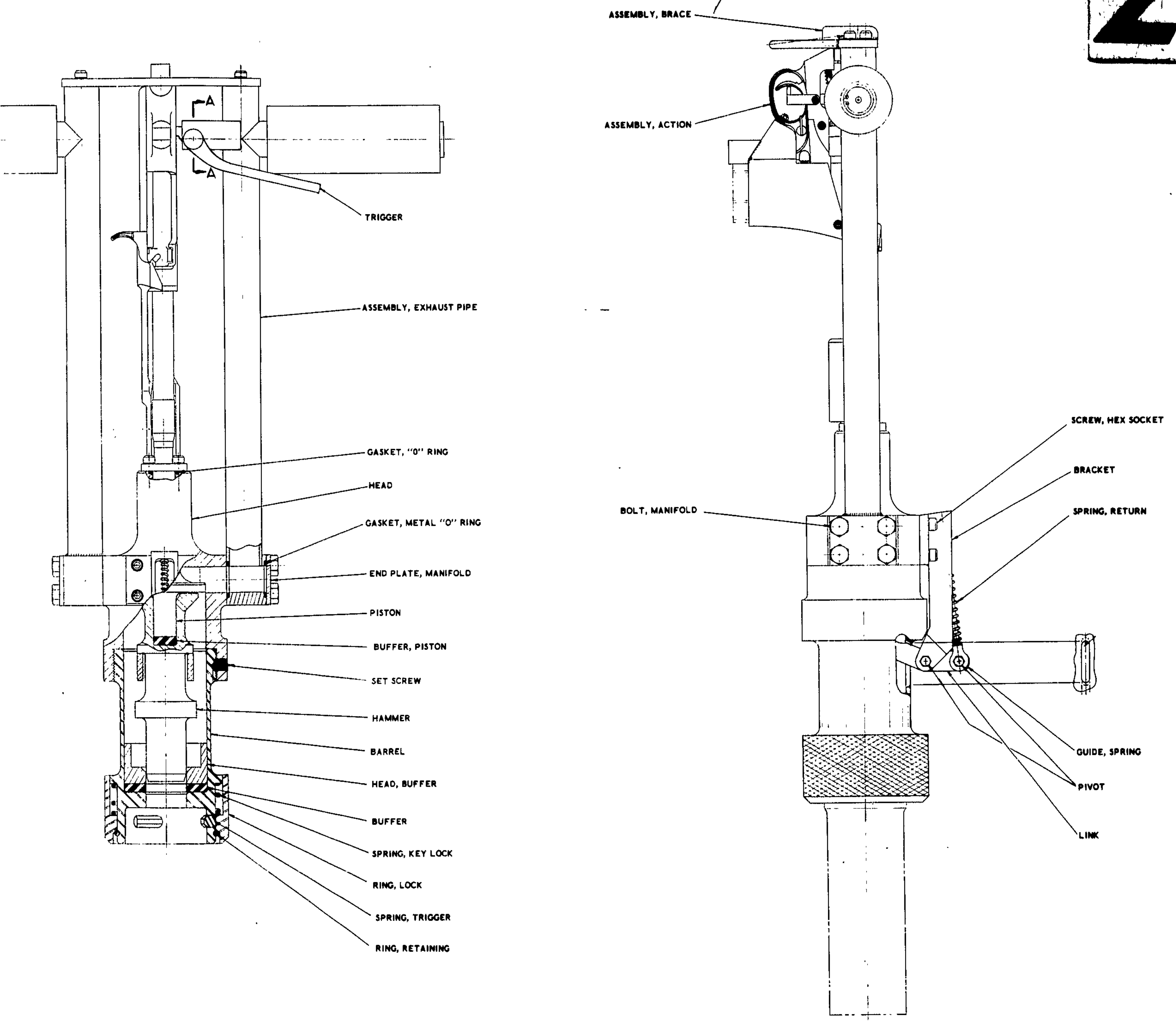
(1) A means to provide rapidly repeating propellant explosions.* This has been satisfied by the incorporation of a gas operated semi-automatic Caliber .30 carbine action which fires Grenade Launcher Cartridges M6 ("Blanks").

*It should be noted that the term "explosion" is not a technical description of propellant ignition; it is used here in a narrative sense.



Assembly Drawing, Ballistic Hammer (Emplacement Mode)

Figure 16



Assembly Drawing, Ballistic Hammer (Emplacement Mode)

Figure 16

(2) A means to return the weight to its starting position so that, upon the occurrence of the next explosion, the associated pressure and energy can again act upon the weight, enabling it to acquire kinetic energy before its next impact with the stake. This has been satisfied by the incorporation of a helical coil spring which pushes the weight back up the chamber after its impact with the stake.

(3) A means for routing the exhaust gases from the chamber. Since the device works on the principle of multiple dynamic loadings, the chamber pressure must be exhausted to enable the spring mentioned above to push the weight up into its original position.

(4) A handle system to enable the operator to position and control the device.

The necessity for incorporating the features mentioned above resulted in the design shown in Figure 16.

The following both physically and functionally describes the major components and subassemblies of the Device. Reference should be made to Figure 16 during this description.

The standard Caliber 30 Carbine Action M1, with the majority of the barrel removed. - Provision is made at the barrel end to adapt the Action to the Head. Three 1/4" - 20 x 3/4" cap screws accomplish this. Provision is also made to restrain the other end of the Action by the Brace in order that the action be rigidly fixed relative to the overall Device. The standard safety which is part of this Action serves as a safety for the tool by locking the Trigger in the "Safe" position. The Device has been designed such that the safety is readily accessible. The function of the safety is as usual; to prevent ignition of a cartridge when it is not desired. As with any ballistic device, the safety should be engaged when the Device is not in use. An addition to the Action is the Magazine Support. This is essentially a long, rectangular box which encases the magazine to protect it during recoil. The Magazine Support is bolted to the Action in two places -- at the trigger guard and the point of attach-

ment of the receiver and the trigger assembly.

The Head - Externally, the Head is essentially a cubical piece of steel with its top and bottom ends machined to cylindrical shapes. Provision is made at the one end to adapt the Action to the Head, as previously mentioned. Internal threads on the other end adapt the Barrel to the Head. A set screw is provided in this piece to prevent rotation of the Barrel with respect to the Head. Internally, the Head has provisions for allowing the propellant gases produced by the explosion of the cartridge to act against the top of the Hammer, provisions for guiding the Hammer in its travel at its smallest diameter, and provisions for allowing the propellant gases to exhaust after the piston has traveled one inch.

The Barrel - This is a hollow, cylindrically shaped steel piece with a shoulder at its bottom end and threads at its top end. The shoulder at the Barrel's bottom end provides a rigid support for the Buffer and Buffer Head. The external thread at the other end enables the Barrel to be mounted to the Head. Provision is made to engage the set screw previously mentioned.

The Buffer and Buffer Head - The Buffer is a disc of butyl rubber with a hole in its center through which the bottom end of the Hammer can pass on its travel to the stake. The Buffer Head is a cylindrical piece of steel which has a shoulder on one end. In normal operation, the Buffer and the Buffer Head serve no purpose. If the Device were actuated with the stake not in place, however, the lower flange of the Hammer would impact the Buffer Head and Buffer, thus cushioning the impact of the Hammer and Barrel. The Barrel would be damaged under these circumstances if it were not for the presence of this shock absorbing system.

The Hammer - This unit is composed of two separate steel pieces, the shapes of which can best be described by reference to Figure 16 . The center lines of the Hammer and Barrel coincide. While in motion along its center line, the Hammer is guided at its largest

diamter by the Barrel and at its smallest diameter by the Head. Note that, upon ignition, the propellant charge exerts its pressure and energy at the Hammer's smallest diameter. After the Hammer travels one inch, it uncovers the scalloped portion of the Head, exhausting the gases. Upon impact with the stake, the bottom end of the Hammer decelerates more quickly than its top end. The rubber cushion between these two ends tends to cushion the Hammer's impact with the stake. This tends to lengthen the time of contact of the Hammer with the stake, and damps the dynamic shock wave which travels through the piston as a result of Hammer-Stake impact.

The Exhaust Manifolds - These are cubical pieces of steel with holes drilled in them such that the flow direction of the incoming exhaust gases is changed by 90°. Also included are bolt holes for connecting the manifolds to the sides of the Head. Affiliated with the Exhaust Manifolds are the Manifold End Plates which are rectangular pieces of steel with holes for the bolts mentioned above. These stop the exhaust gases from escaping to the outside. Two Manifold Gaskets seal each Exhaust Manifold from exhaust leakage between the Head and Manifold End Plates. It should be pointed out that each Exhaust Manifold is welded to an Exhaust Pipe-Handle. Each Exhaust-Pipe Handle Assembly consists of two pieces of thin-walled 1-1/8" O.D. steel tubing, which are welded together to form the shape of an inverted "L". The lower end of each inverted "L" is welded to its respective Exhaust Manifold. The other end has a plug welded in to accommodate the cap screws fastening the Brace to the Assembly. The remaining horizontal bar of each "L" is the handle with which the operator guides the Device. At the open end of each Handle is a Muffler to silence the propellant ignition. Each Muffler consists of a steel cylinder with a small hole drilled through the center. These are pressed into their respective Handles and are each held in place with a retaining ring. Notice that the exhaust gases travel from the Head, to the Exhaust Manifolds, where their direction is changed by 90°, up to the Exhaust Pipes, and out through the Handles.

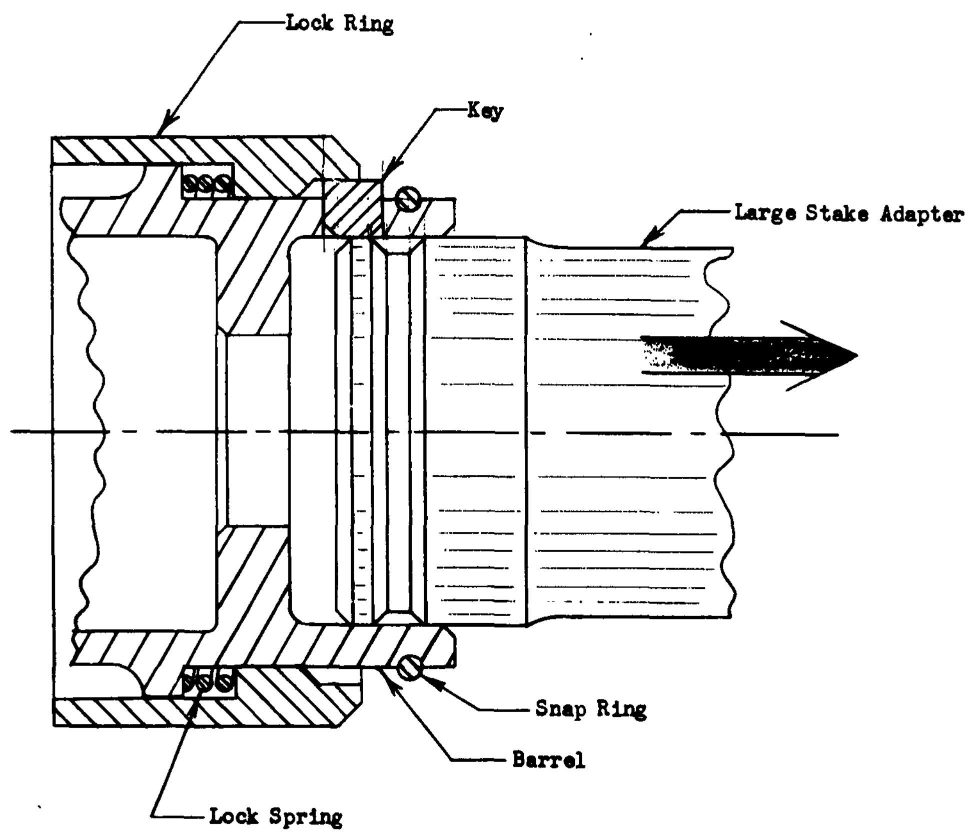
The Brace - This is a rectangular piece of steel plate. A square hole under the small piece of steel bar welded to the top of the Brace adapts the caliber 30 action to it. Notice the semi-circular, round steel rod welded to the side of the Brace. The purpose of this rod is to protect the trigger assembly of the action should the Device fall over. The Brace is attached to the Exhaust Pipes by four 1/4" - 20 x 3/4" cap screws which screw into the plugs welded in each Exhaust Pipe.

The Trigger - This is a piece of sheet steel formed into a channel shape and pivoted on a block welded to an Exhaust Pipe. There is a square hole in one side of the channel. This accommodates the Trigger Spring, which is a torsion bar, one end of which is attached to the Trigger by the above square hole. The other end of the Trigger Spring is attached to the block welded to the Exhaust Pipe. The Trigger extends out to contact the Trigger Actuator. This is a piece of rectangular steel bar which is formed into a "U" shape. It is pivoted through both legs on the same pin as the Trigger of the caliber 30 action. Pulling the Trigger causes one end of the Trigger Actuator to depress and, because of the pivot, the other end rises, which actuates the trigger of the caliber 30 action.

The Key System - In the end of the Barrel farthest from the caliber 30 action are three rectangular holes, each approximately 1-1/4 inches long. Three curved Keys, each having a square cross-section with beveled edges, fit into these rectangular holes. In the usual operation these Keys mesh with a groove in the Stake Adaptors. As can be seen in Figure 16, this Adaptor cannot be moved by pulling or pushing, since this radially pushes the Keys tight against the Lock Ring. The Lock Ring is a hollow, cylindrical piece of steel. It is restrained from movement in one direction by the Snap Ring, which engages a semi-circular groove in the Barrel. Its movement in the other direction is restrained by the helical Lock Spring only, enabling a total movement of the Lock Ring of

approximately 1/2 inch. If this is done and the Adaptor is pulled upon, the Keys will move radially outward into the space created by the Lock Ring movement. (See Figure 17). The Adaptor can then be removed from the Device.

The Hammer Return Assembly - This Assembly consists of four components; the Link, Bracket, Hammer Return Spring, and Spring Guide. The Bracket has the shape of an open rectangular box with a hole through the sides at one end to provide a pivot for the Link. Another hole is provided at the other end perpendicular to this. The tubular end of the Spring Guide pases through this hole during the Guide's travel. Notice the four holes provided for four 1/4" - 20 x 3/4" cap screws which secure the Bracket to the Head. The Link is a piece of steel, with the shape of a yoke. A hole is provided near its middle to connect the Link and the Bracket, using the Link Pin. At one end of the Link is a hole to enable the Link and Spring Guide to be pin-connected. The other yoked end of the Link fits under the shoulder on the Hammer. This is the means by which the Link engages the Hammer to push it up to its original position. The Spring Guide consists of a slender steel tube with a small yoke on one end. The yoke end of the Guide engages the hole at the end of the Link. The two components are connected with the Guide Pin. The other end passes through that hole in the Bracket which is perpendicular to the hole about which the Link pivots. A shoulder is provided on the yoke end of the Guide to accommodate the Hammer Return Spring, a long, slender helical compression spring. One end of the Spring contacts the Bracket while the other contacts the shoulder on the Guide. The operation of the overall Hammer Return Assembly is as follows: Before ignition of the cartridge, the Hammer is held in its extreme upward position by the preload in the Spring. Upon ignition, the Hammer begins to move down toward the Stake. This motion causes the yoke end of the Link which pushes against the shoulder of the Hammer to move down also. This, in turn, causes the end of the Link which is connected to the Guide to move up, since the Link is pivoted near its



KEY SYSTEM OPERATION DURING REMOVAL
OF LARGE STAKE ADAPTER

center at the Bracket. This Guide motion relative to the Bracket causes the Spring to compress farther. When the Hammer eventually impacts the Stake, its energy is completely dissipated and downward motion ceases. The fully compressed spring then pushes against the Bracket and Guide which moves the Guide end of the Link down. Because of the Bracket pivot, the Hammer end of the Link moves up, pushing the Hammer up to its original position. The cycle is repeated when the next cartridge is initiated.

The Large Stake Adaptor - This consists of a tubular member constructed of aluminum. There are grooves in one end of the Adaptor. This, in conjunction with the Key System, is the means by which the Adaptor is attached to the Device. The purpose of the Adaptor is to furnish a means of guiding the tool with respect to Ground Stake GP 113/G during emplacement. The Adaptor has been designed such that, before firing, the top of the Stake head butts against the lower side of the shoulder in the Barrel of the Device. The Adaptor, mounted on Ground Stake GP 113/G is seen on the following page.

The Small Stake Adaptor - This encompasses every feature of the Large Stake Adaptor, except its configuration has been altered to accommodate Stake GP 112/G. This Adaptor, mounted on Ground Stake GP 112/G, is seen on the following page.

c. Operating cycle

The previous discussion was concerned with the physcial and functional description of each component and subassembly of the Device. The following discussion is concerned with their interrelation in the operation of the overall Device.

The operating cycle of the Device is as follows: (Refer to Figure 16). Upon ignition of the propellant charge (standard Caliber 30 grenade cartridge) by actuation of Trigger on the semi-automatic Caliber 30 carbine action, the pressure and energy of the burning propellant gases acts upon the top of the Hammer. This causes the Hammer to move toward the stake. Notice that at the start of its motion, the Hammer begins to compress the small diameter helical coil spring because of the

Ground Stake GP-113/G Adaptor

Figure 18

Ground Stake GP-112/G Adaptor

Figure 19

movement of the connecting link system. After the piston has traveled 1 inch, it will have reached its design velocity. At this point the propellant gases are exhausted. This occurs when the top end of the Hammer uncovers the scalloped portion of the Head. (See Figure 20). This routes the exhaust gases out through the Exhaust Pipes and Handles. After this occurs, the Hammer impacts the stake. This drives the stake a certain distance into the ground. The Hammer continues to drive the stake into the ground until they either lose contact or the Hammer's kinetic energy is decreased to nearly zero. At this point, the helical compression spring is fully compressed, and through the link system, it pushes the Hammer to its initial starting position.

While the above was occurring, the pressure in the barrel of the carbine action was sufficient to enable it to extract and eject the spent case and strip another cartridge from the magazine into its receiver. When the Trigger of the action is again actuated by the operator, the operating cycle will occur again.

d. Interior ballistics

(1) Analytical development

Certain desired design parameters for the impact hammer will prescribe the associated interior ballistics. These parameters include:

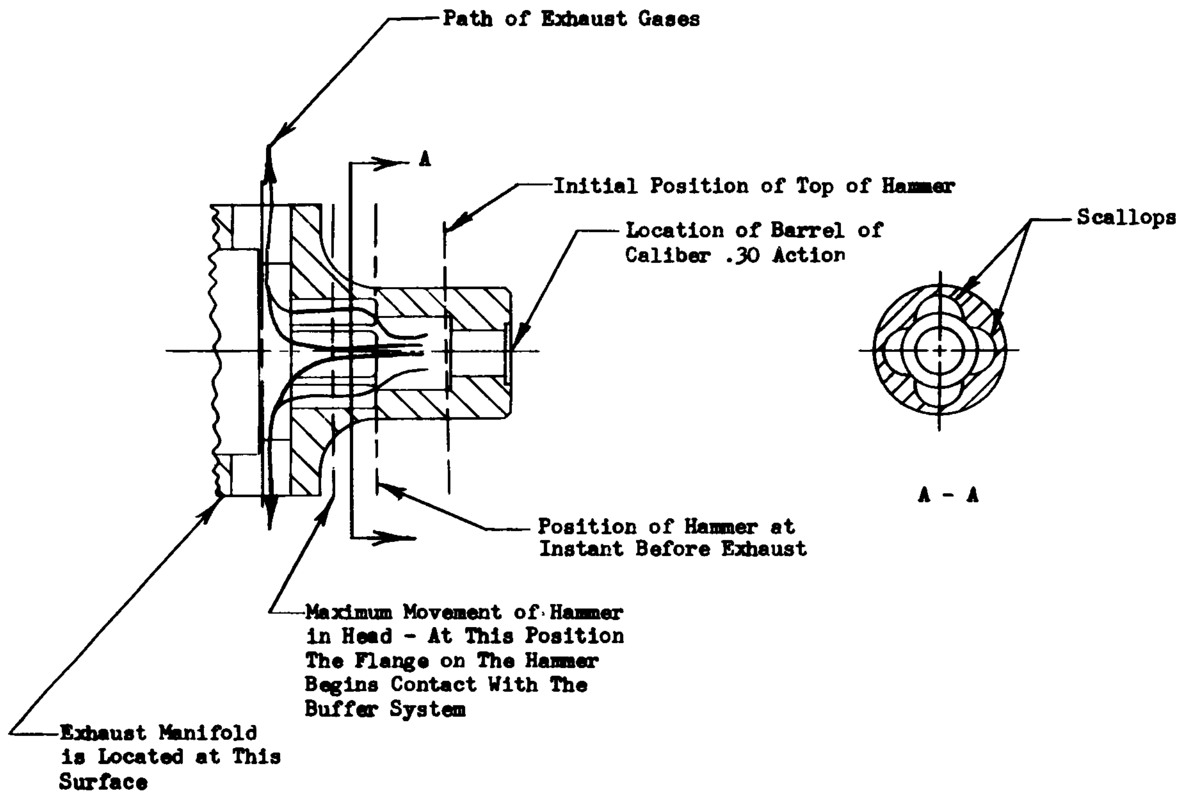
(a) Hammer energy required

(b) Maximum allowable pressure

(c) Maximum desired exhaust pressure

(d) Piston area

The piston energy is derived thermodynamically from extremely hot propellant gases. The magnitude of pressure created in a fixed volume by a given propellant is described by the propellant impetus as;



SECTIONS THROUGH HEAD ILLUSTRATING GAS EXHAUSTION

Figure 20

$$P_1 = \frac{12\ CF}{V_1} \tag{1}$$

Where C = charge weight; lbs.

F = propellant impetus; ft-lb/lb.

P_1 = initial pressure; lb/in^2

V_1 = initial volume; $in.^3$

As the gas expands from one state to another, assuming an adiabatic relation, the work done is;

$$W = \frac{P_1 V_1 - PV}{12\ (\gamma - 1)} \tag{2}$$

where P = pressure at any volume, V, lbs/in^2

V = second state of expansion; in^3

P_1 = initial pressure; lb/in^2

V_1 = initial volume; in^3

W = work output; ft-lbs.

γ = ratio of specific heats of propellant gases

From the adiabatic relation;

$$P = P_1 \left[\left(\frac{V_1}{V}\right)\right]^{\gamma} \tag{3}$$

Coming equations (2) and (3)

$$W = \frac{P_1 V_1 \left[1 - \left(\frac{V_1}{V}\right)^{\gamma - 1} \right]}{12(\gamma - 1)} \tag{4}$$

Substituting CF for $P_1 V_1$ and describing the work done as the kinetic energy of the hammer;

$$1/2\ mv^2 = \frac{CF \left[1 - \left(\frac{V_1}{V}\right)^{\gamma - 1} \right]}{\gamma - 1} \tag{5}$$

where:

m = mass of hammer; slugs

V = velocity of hammer; ft/sec.

One final relation is the volume expansion. Part is due to the piston movement and part due to the hammer itself as it recoils.

$$V = V_1 + A(X + X_H) \tag{6}$$

where:

A	=	piston area	in^2
X	=	piston movement	in.
X_H	=	hammer movement	in.

From the conservation of momentum, the velocity of the hammer will be related to the piston by;

$$m_H\ v_H = m_P\ v_P \tag{7}$$

$$\therefore X_H = \frac{m_P}{m_H} X$$

Then eqūation (6) can be written

$$V = V_1 + A\left(X + \frac{m_P X}{m_H}\right)$$

$$V = V_1 + AX\left(1 + \frac{m_P}{m_H}\right)$$

$$V = V_1 + \left(1 + \frac{m_P}{m_H}\right) AX \qquad (8)$$

Equations (5) and (8) relate the derived piston energy from a given charge weight at a given Hammer stroke, X, which is the travel of the Hammer relative to the Earth. The results of the Interior Ballistics calculations can be found in the table on the following page. These facts are pertinent to the calculations;

(1) The driven diameter of the Hammer is one inch.

(2) The initial volume (V_1) = .50 in^3

Using the standard carbine grenade launcher cartridge as the power source provides 20 grains of propellant. The impetus is estimated as F = 300,000 ft-lb/lb., corresponding to a peak pressure of 20,600 psi in the locked shut condition. This pressure seldom occurs as the piston usually moves creating a larger volume when all the propellant is burned. None of the energy is lost by assuming all burning occurs in the initial volume, but the actual peak pressure is something less than that indicated at x = o. Explosive ordnance experience shows that this type propellant usually burns in .2 to .3 millisecond and from the pressure travel curve, a fair estimate of the peak pressure can be obtained on the adiabatic expansion curve.

Increments of time are obtained over small travel increments using the mean velocity over that increment. the recoiling mass

1	2	3	4	5	6	7	8	9	10	11	12	13
x (in.)	1.138AX (in^3)	V (in^3)	V/V_1	$(4)^{.25}$	1/ (5)	1- (6)	$\frac{CF}{\gamma - 1}$ (7) (ft-lb)	v (ft/sec)	$(4)^{1.25}$	P (psi)	Δt (millisec)	t (millisec)
.1	.0894	.589	1.178	1.042	.960	.040	137	47	1.227	20,600 16,800	.307	.307
.2	.1788	.679	1.358	1.080	.926	.074	254	64	1.465	14,000	.130	.437
.4	.3575	.858	1.716	1.144	.874	.126	432	83.5	1.962	10,500	.306	.743
.6	.5360	1.036	2.072	1.200	.833	.167	572	96	2.485	8,300	.255	.998
.8	.7150	1.215	2.430	1.248	.801	.199	682	105	3.035	6,800	.224	1.222
1.0	.8940	1.340	2.680	1.280	.781	.219	750	110	3.430	6,000	.210	1.432

TABLE OF CALCULATIONS

Figure 21

weighs 28.5 pounds (*) and the Hammer, 4 pounds.

This data is plotted in Figure 22 with the estimated time pressure trace.

(2) Experimental Results

The preceding analytical investigation provided a background for establishing the design configuration of the Device. In order to optimize it with respect to tolerable recoil and penetration capability, an extensive experimental program was initiated. This revealed that exhaust should occur after a Hammer motion relative to the Head of one inch (Refer to the Design Layout). Under this condition, the penetration per blow was a maximum and the associated recoil impulse was at a level which could be comfortably tolerated by the operator over a prolonged operating period. Under this condition,

$$X_H + X_p = 1.00$$

$$X_p \left(1 + \frac{mp}{mH}\right) = 1.00$$

$$X_p = X = .88$$

Reference to the Table of Calculation indicates that this (X) corresponds to a piston velocity (interpolated) of 107 ft/sec. The maximum gas pressure is approximately 16,500 psi.

In order to verify the analytical velocity determination, high speed movies were taken of the Guide Rod of the Hammer Return Assembly while the Device was in operation. From this data, the velocity of the Hammer was found to be 116 ft/sec. This corresponds very closely to the analytical determination of this velocity.

Corresponding to 116 ft/sec., the momentum of the 4 pound Hammer is:

(*)This weight was obtained with the GP 112/G Adaptor installed on the Device. It is an analytical weight.

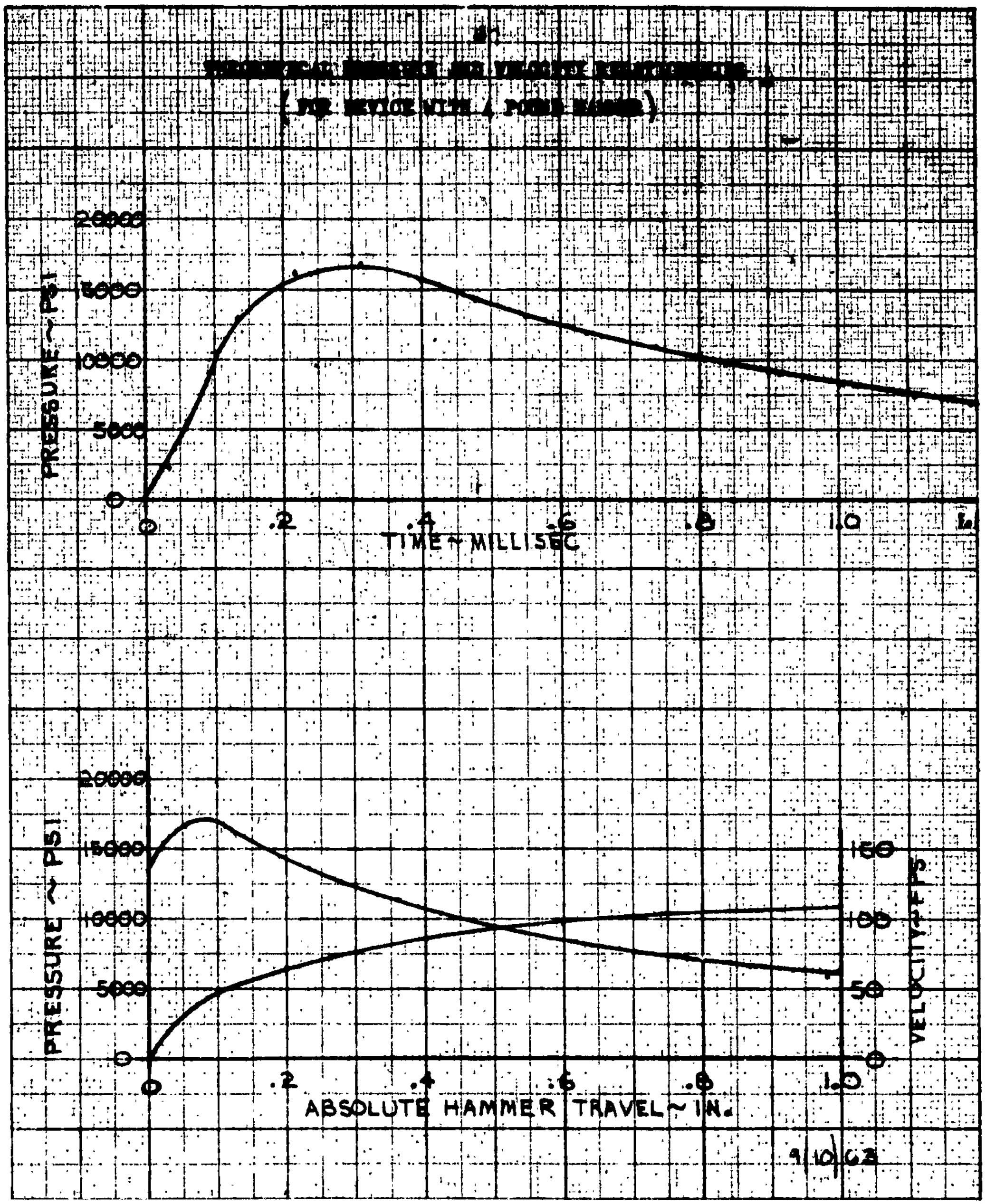

20000
15000
10000
5000
0
PRESSURE ~ PSI
0
.2
.4
.6
.8
1.0
TIME ~ MILLISEC
20000
15000
10000
5000
0
PRESSURE ~ PSI
150
100
50
0
VELOCITY ~ FPS
0
.2
.4
.6
.8
1.0
ABSOLUTE HAMMER TRAVEL ~ IN.
9/10/62

Figure 22

$$I = m\,V$$

$$= \frac{4}{32.2}\ (116)$$

$$I = 14.5 \text{ lb-sec.}$$

The corresponding kinetic energy per blow is:

$$K = 1/2 \ m\,V^2$$

$$= 1/2 \ \frac{(\ 4\ )}{32.2}\ 116^2$$

$$K = 835 \text{ ft-lb.}$$

After the design velocity had been attained, pressure-time data were desired for the Device. For this phase, a mockup 3 pound piston and a mockup Head which employed the modified caliber 30 action firing grenade launcher cartridges were used. (In short, the mockup system was identical to the Device from an Interior Ballistics standpoint, except for the 3 lb. piston). The results of this investigation are shown on the following page. The maximum pressure was found to be 17,000 psi, compared to an expected maximum of 16,800 psi. (See Page 4.48). It should be pointed out that while the analytical and experimental maximum pressures agree, the shapes of the analytical and experimental pressure-time curves appear to be different. This is because the analytical pressure-time curve was assumed at the point of ignition. The experimental results, however, incorporated a time lag while the piston moved before uncovering the pressure tap. From time = 0 to time = .3 milliseconds the piston, because of its inertia, began slowly uncovering the pressure tap. As it uncovered more and more of the tap, increased amounts of entering gases provided an indication of a rapidly rising pressure coinciding with the predicted p-t curve.

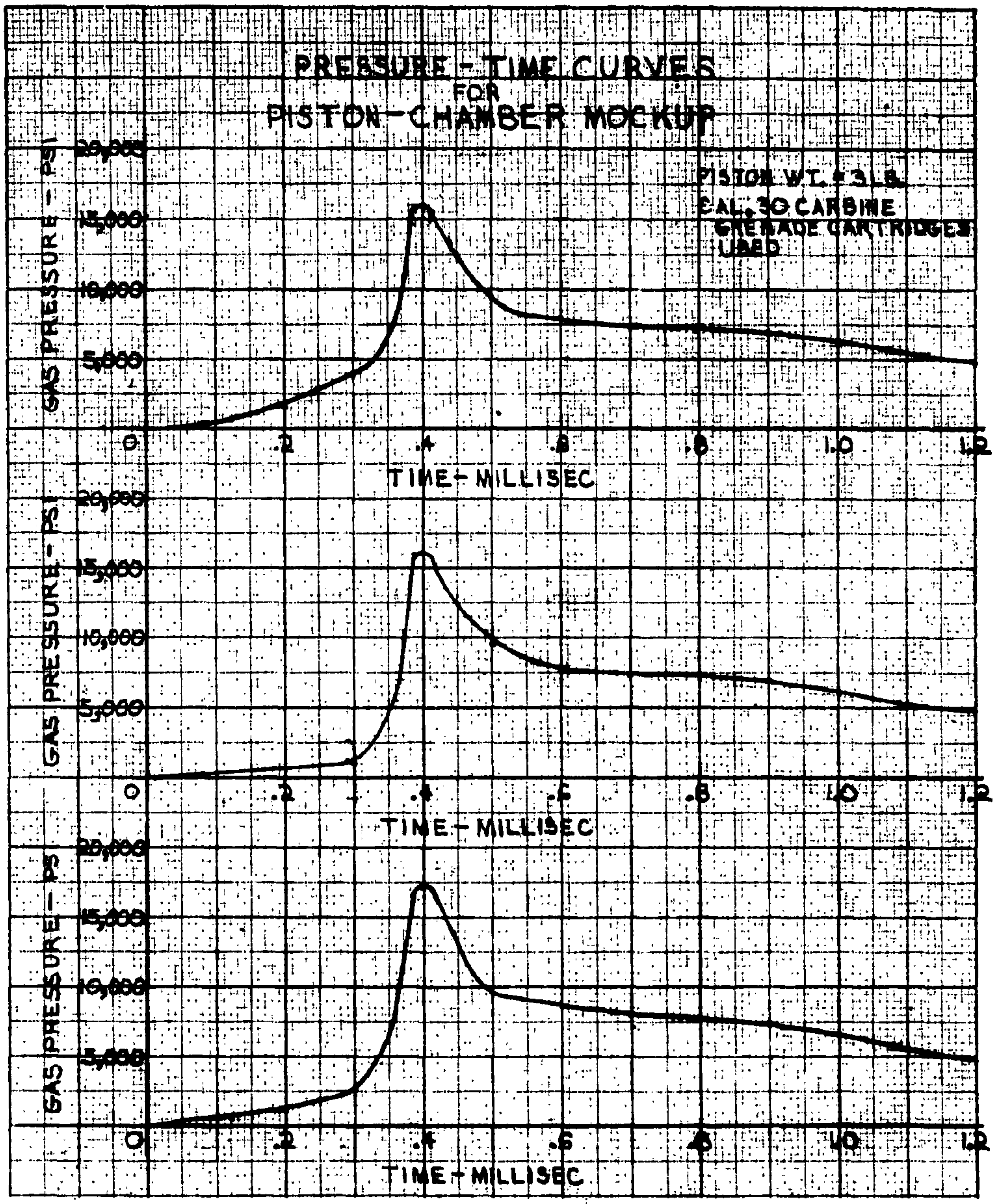

PRESSURE - TIME CURVES
FOR
PISTON - CHAMBER MOCKUP
PISTON WT. = 3 LB.
CAL. 30 CARBINE
GRENADE CARTRIDGES
USED
GAS PRESSURE - PSI
20,000
15,000
10,000
5,000
0
.2
.4
.6
.8
1.0
1.2
TIME - MILLISEC

Figure 23

The Device actually employs a 4 pound Hammer, instead of 3 pounds used in the experiments. If this test were repreated with identical conditions, except for the Hammer weight, the peak pressure would be slightly less because of the increase in work required by the gases to move the Hammer. The pressure-time relation to the right of this peak would be slightly higher since a higher pressure force would be required to move the Hammer out of the Head.

(3) Available Ammunition

The standard caliber 30 M6 grenade launcher cartridge is used as the power source of the Device. Since these cartridges are standard military items, no problems of any nature are foreseen regarding their availability.

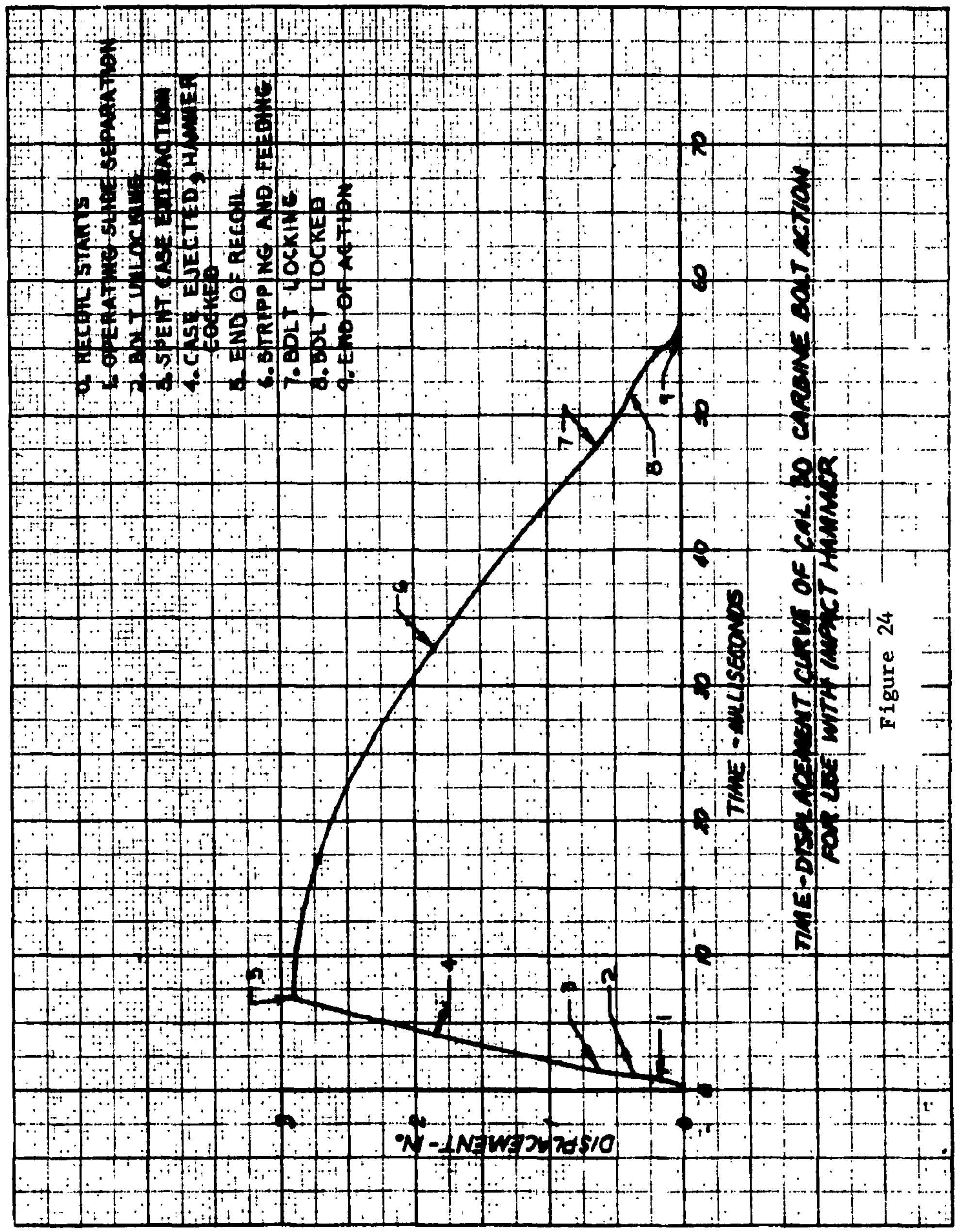
d. Kinematic Analyses

(1) Modified caliber 30 carbine action cycle

In the beginning stages of the development of the Ballistic Hammer, a kinematic analysis was conducted on the carbine mechanism's cyclic action. This was done to ensure that the mechanism began extraction of the spent cartridge case after the exhaustion of the high pressure propellant gases. Otherwise, serious damage to the action could result.

The complete kinematic analyses is found in AAI Report, No. ER-2937, pp 31 to 55. It is not considered necessary to present it in its entirety here.

The summary of the results of this investigation is discussed in the following paragraphs. From these calculations, it can be seen that the opening of the bolt is completed (Sta. 2) 1.50 milliseconds after the firing of the cartridge. It was calculated that 1.30 milliseconds after the first cartridge is fired the Hammer uncovers the exhaust ports. This allows the exhaust gases to expand into the atmosphere, which drops the Head pressure to a safe extraction level.



TIME-DISPLACEMENT CURVE OF CAL. 30 CARBINE BOLT ACTION FOR USE WITH IMPACT HAMMER

Figure 24

In this analysis, an intial propellant pressure of 16,000 psi was assumed to act on the recoil piston of the carbine mechanism. In the actual mechanism, however, the gases must travel through a nozzle before acting on this piston, with a considerable corresponding pressure drop. This pressure drop in the actual mechanism increased the calculated cycling time of the mechanism. Also, friction in the mechanism was neglected (with the exception of that required to strip the cartridge from the magazine). This friction also tended to increase the calculated cycling time of the mechanism.

On the basis of the analysis, it can be stated that the calculated cycle time is low compared to the actual cycling time and that, in the actual system, the Hammer of the Device will uncover the exhaust ports well before the uncamming of the bolt begins.

The above was verified experimentally from high speed movie data taken of the Device while in operation.

(2) Hammer cycle

At the end of the full power stroke the Hammer leaves the drive tube permitting the gas to expand and push with a much reduced pressure. Opposing this motion is a helical coil spring whose function is to return the Hammer and assure a firm seat of it when the cartridge is fired. After impacting the stake, the remaining energy is absorbed by rubber buffers with low elastic properties. The Hammer is returned by the coil spring.

The propellant gases accelerate the Hammer to an intial velocity $\dot{X}_o$, and then exhaust to a reduced pressure. The inertia of the Hammer enables it to further drive against the recoil spring for a small distance before the Hammer impacts the stake, which further reduces its velocity.

During the deceleration against the spring:

$$m\ddot{x} = -(F_o + kx) \qquad (1)$$

where

m	=	mass of Hammer	slugs
F_o	=	spring preload	lbs.
k	=	spring rate	lb/in
x	=	Hammer travel	in.
$\dot{x}$	=	Hammer velocity	ft/sec.
$\ddot{x}$	=	Hammer acceleration	ft/sec.^{2}

The solution for the velocity as a function of travel is:

$$\dot{x}^2 = \dot{x}_o^{\,2} - \frac{2}{m}\left[F_o x + \frac{k}{2} x^2\right] \tag{2}$$

The time involved by integrating equation (2) is:

$$t = \frac{1}{\sqrt{k/m}}\left[\sin^{-1}\frac{F_o + kx}{\sqrt{x_o^{\,2}\, mk + F_o^{\,2}}} - \sin^{-1}\frac{F_o}{\sqrt{x_o^{\,2}\, mk + F_o^{\,2}}}\right] \tag{3}$$

After impact the velocity is reduced according to the conservation of momentum and some assumed coefficient of restitution, e. By definition:

$$m_1 u_1 + m_2 u_2 = m_1 v_1 + m_2 v_2 \tag{4}$$

$$e = \frac{v_2 - v_1}{u_1 - u_2} \tag{5}$$

In terms of this coefficient the velocities after impact are;

$$v_1 = u_1 - (1 + e) \frac{m_2 (u_1 - u_2)}{m_1 + m_2} \tag{6}$$

$$v_2 = u_2 + (1 + e) \frac{m_1 (u_1 - u_2)}{m_1 + m_2} \tag{7}$$

where

m_1 = mass of Hammer -- slugs

m_2 = mass of ground stake --slugs

u_1 = Hammer velocity before impact -- ft/sec

u_2 = stake velocity before impact $(u_2=0)$ -- ft/sec

v_1 = Hammer velocity after impact -- ft/sec

v_2 = stake velocity after impact -- ft/sec

e = coefficient of restitution

After striking the stake the Hammer is stopped by the rubber buffer within a small travel. The return velocity is determined from;

$$m \ddot{x} = F_3 - kx \tag{8}$$

Integrating with $\dot{x}_3 = 0$

$$\dot{x}^2 = \frac{2}{m} \left[F_3 x - 1/2\ kx^2 \right] \tag{9}$$

Solving for the time involved:

$$t = \frac{1}{\sqrt{k/m}} \left[\sin^{-1} \left(\frac{kx}{F_3} - 1\right) + \frac{\pi}{2} \right] \tag{10}$$

Equations (1) through (10) describe the motion of the Hammer and cyclic time during each increment of travel.

The Device uses a 4 pound Hammer with a design velocity of 116 ft/sec. acting against the following spring:

$$F_o = \frac{.37\ (33.1)}{2}$$

$$F_o = 6.2 \text{ lb.}$$

$$K = \frac{33.1}{2} \text{ lb/in}$$

$$K = 16.6 \text{ lb/in}$$

The impact velocity of the Hammer against the Stakes after a total of one inch of relative travel is 116 ft/sec., since, in this case, the Hammer reaches its design velocity at the instant that it impacts the Stake.

The time to travel the one inch is approximately

$$t = \frac{X}{12\ V}$$

$$= \frac{1}{12(116)}$$

$$t = .71\ X10^{-3} \text{ seconds}$$

For Ground Stake GP 113/G which weighs 13 pounds the Hammer velocity after impact is determined from equation (6) with a coefficient of restitution (e) of .6;

$$V_1 = \mathcal{U}_1 - (1 + e) \frac{m_2 \quad (\mathcal{U}_1 + \mathcal{U}_2)}{m_1 + m_2}$$

$$= 116 - 1.6 \quad \left(\frac{13}{13 + 5}\right) \quad (116-0)$$

$$V_1 = -18 \text{ ft/sec.}$$

The negative velocity indicates a change in direction after impact and infers that the Buffer System is not required. However, it is included in the design to ensure safety to the Barrel structure in the case of a "no load" firing. Neglecting the rebound velocity, the maximum return velocity is determined from equation (9).

$$\dot{x}^2 = \frac{2}{m} \left[F_3 X - 1/2 \, KX^2 \right]$$

$$= \frac{2(32.2)}{4} \left[\frac{22.8 \, (1)}{12} - \frac{1}{2} \frac{16.6 \, (1)^2}{12} \right]$$

$$\dot{x}^2 = 19$$

$$\dot{x} = 4.4 \text{ ft/sec.}$$

The time involved is approximately:

$$t = \frac{2X}{V}$$

$$= \frac{2 \, (1)}{12 \quad (4.4)}$$

$$t = .038 \text{ seconds.}$$

The time required for the Hammer to return to its initial position is approximately .039 seconds. This is a sufficiently high

response time, since, after firing, the operator's response to pull the trigger requires more time than the above.

f. Stress Analysis

The following analysis substantiates sufficient factors of safety on all components in critical, highly stressed areas of the Emplacement Device. Each part is presented with the type material and material properties. Endurance limits in shear and bending are assumed as a fraction of the ultimate stress.[16] The endurance limit in shear is assumed as 0.53 the endurance limit in bending, and the endurance limit in direct stress is assumed as .85 the endurance limit in bending. When the endurance limit in bending is not given, it is assumed as 0.44 the ultimate stress.

The general design arrangement is shown in Figure 16.

(1) Head (Type 416 C R Steel)

(a) As indicated in the Interior Ballistics section, a maximum pressure of 20,600 lb/in^2 can be expected in the locked shut condition. Assuming this pressure, the maximum stress is found to be:

$$S = P\left(\frac{b^2 + a^2}{b^2 - a^2}\right)$$

$$= 20{,}600 \quad \frac{(.875^2 + .500^2)}{.875^2 - .500^2}$$

$$S = 38{,}100 \ lb/in^2$$

$$\text{Endurance limit} = .85 \times .44 \times S_u$$

$$= .85 \times .44 \times 185{,}000$$

$$= 69{,}200 \ lb/in^2$$

The factor of safety for infinite life based on this endurance limit is:

$$F.S. = \frac{S_e}{S}$$

$$= \frac{69200}{38100}$$

$$F.S. = 1.8$$

(b) The 20,600 lb./in.^{2} pressure mentioned above tends to pull the action from the Head. The force associated with this pressure is:

$$F = p\ A$$

$$= 20{,}600\ (.75)^2 \left[\frac{\pi}{4}\right]$$

$$F = 9100 \text{ lb.}$$

Resisting this force are three 1/4" - 20 x 3/4" cap screws, each with a minimum breaking strength of 4,950 lb.:

$$FS = \frac{3(4{,}950)}{9100}$$

$$FS = 1.63$$

(2) Hammer (4340 Steel)

The Hammer strikes Ground Stake GP 113/G with sufficient force to peen the head of this Stake. This means that the yield bearing strength of the head is exceeded. The force required to cause peening is:

$$F_1 = S_{bry}\ (A_s\ )$$

$$= (75{,}000)\left[\frac{\pi}{4}\right]\ (1.19)^2$$

$$F_1 = 83{,}500 \text{ lb.}$$

This force exists in the tip of the Hammer and in the Head of the Stake.

The corresponding maximum stress in the Hammer is:

$$Sm = \frac{F_1}{A_p} = \frac{83,500 \ (4)}{\pi \ (1.32)^2}$$

$$= 63,000 \text{ psi}$$

The tensile strength of the 4130 steel is 185,000 psi., indicating an endurance limit of;

$$Se = .44 \ (.85) \ Su$$

$$= .44 \ (.85) \ 185,000$$

$$Se = 69,400 \text{ psi.}$$

The factor of safety for an infinite life based on this endurance limit is:

$$F.S. = \frac{Se}{S}$$

$$= \frac{69,400}{63,000}$$

$$F.S. = 1.1$$

Further calculations have shown when Ground Stake GP 112/G is emplaced a resultant stress level of one half of that above is encountered.

(3) Exhaust Pipes & Handles (Type 304 C R Steel)

(a) Stresses due to gas pressure

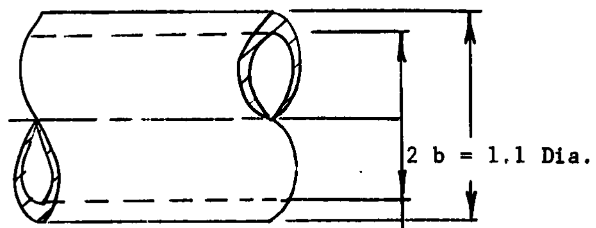
Assuming an isothermal expansion into the large low pressure volume, the maximum pressure which can exist in the Pipe is found to be:

$$P_2 = \frac{P_1V_1}{V_2} = \frac{20{,}600 \times .495}{(.495 + 19.82)}$$

$$P_2 = 510 \text{ lb/in}^2$$

The corresponding maximum stress is:

$$S_{max} = P_2 \frac{b^2 + a^2}{b^2 - a^2}$$



EXHAUST PIPE

$$S_m = 510 \frac{(.55)^2 + (.45)^2}{(.55)^2 - (.45)^2}$$

$$S = 2580 \text{ lb/in}^2$$

This is not considerable since the yield strength of the material is 30,000 psi.

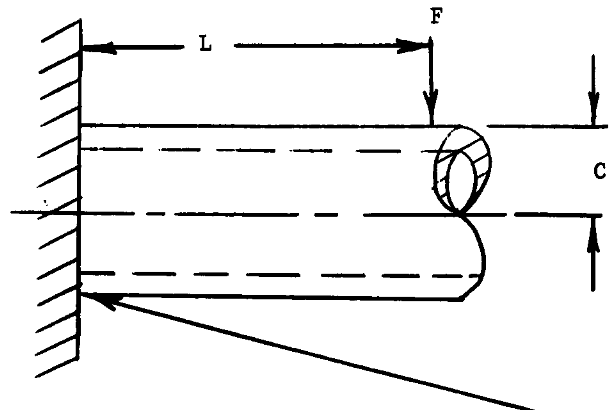
(b) Maximum bending force permitted $= F_m$

Since the exhaust handles are subject to severe abuse during transportation and handling, an estimate is made of the maximum load that can be safely withstood.

$$\frac{Sy}{F:S.} = \frac{FLc}{I}$$

$$F = \frac{30{,}000 \ (\pi) \ .216 \ (.08)}{3.36 \ (1.5)}$$

$$F = 400 \text{ lb.}$$



(4) Barrel (Type 416 C R Steel)

(a) Stress in walls cause by "no load" firing.

If the Device is actuated with a Stake not in place, thè kinetic energy of the Hammer must be absorbed by the Barrel. This is the most severe circumstance which the Barrel could encounter. In this event, the Hammer impacts a Buffer System which decreases the maximum force in the Barrel resulting from this impact. The magnitude of this force is extremely difficult to analytically determine. For this reason, a strain gage investigation was conducted on the Barrel to determine the stress caused when the above occurs.

This investigation revealed that when the "no load" condition occurred, a maximum stress of 68,000 psi was realized by the walls of the Barrel. The corresponding force in the Barrel was:

$$F = SA$$

$$= 68,000 \left[\left(\frac{\pi}{4}\right) (2.9^2 - 2.65^2) \right]$$

$$F = 73,000 \text{ lb.}$$

The 68,000 psi stress is below the yield strength of the steel used (140,000 psi). The yield strength was used as the strength criteria because the "no load" condition is one which occurs only occasionally.

(b) Stress in threads caused by "no load" firing

$$S_s = \frac{F}{A_s}$$

$$= \frac{73,000}{\pi (3.35) 1 (.5)}$$

$$S_s = 13,900 \text{ psi}$$

The "no load" firing condition produces an identical shear stress in the corresponding threads in the Head. In addition to this stress, however, there is a hoop stress in the Head caused by the ramp action of the threads.

The total hoop force is found to be:

$$F_H = F \cot \left(\frac{X}{2}\right)$$

Where X is the included angle of the thread = 60°

$$= 73,000 (1.73)$$

$$F_H = 126,000 \text{ lb.}$$

This force is distributed radially over the threaded portion of the Head.

$$P_R = \frac{F_H}{A_H}$$

$$= \frac{126{,}000}{\pi\ (3.35)\ 1\ (.65)}$$

$$P_R = 18{,}300 \text{ psi}$$

At this pressure the maximum stress is:

$$S = P_H \frac{(b^2 + a^2)}{(b^2 - a^2)}$$

$$= 18{,}300 \frac{(2^2 + 1.7^2)}{(2^2 - 1.7^2)}$$

$$S = 116{,}000 \text{ psi}$$

This stress is below the yield stress of the Head material.

(5) Springs

Other areas of significant stress include the Hammer Return Spring and the Trigger Spring:

(a) Trigger Spring

This spring is a torsion bar with one end being fixed to the Trigger and the other being fixed to the Trigger Mounting Block. A relative motion of 10° occurs between the two ends.

The maximum shearing stress in this torsion bar is:

$$S_s = \frac{\theta\ G\ r_1}{\ell}$$

$$S_s = \left[10^\circ \ \frac{(\frac{\pi}{2})}{90^\circ} \right] \frac{7.3 \times 10^6 \quad (.062)}{1}$$

$$S_s = 79{,}000 \text{ psi}$$

The yield shearing stress for this material is 90,000 psi. This corresponds to a safety factor of:

$$F.S. = \frac{90{,}000}{79{,}000}$$

$$F.S. = 1.14$$

(b) Hammer Return Spring

This spring operates at one-half of the Hammer velocity and has the following design characteristics:

NS 355 (Stainless Steel) wire

O.D. = .281

I.D. = .161

d = .059

Free Length = 4.37

No. Coils = 46 Total (sqd & grd. ends)

Installed Length = 4"

Length at end of stroke = 3.31

Velocity = 60 ft/sec.

The total deflection of this spring is divided into two parts -- static and dynamic. From the above, the static deflection is found to be:

$$X_s = 4.37 - 3.31$$

$$X_s = 1.06 \text{ inches}$$

The total dynamic deflection is:

$$X_D = \frac{VoL}{N\ (88{,}600)\ \left(\frac{d}{D}\right)}$$

$$= \frac{60\ (12) \left[\pi\ (44)\ .22\right]}{(88{,}600)\ (.059)} \quad .22$$

$$X_D = .92 \text{ inches}$$

The total deflection of the spring is therefore:

$$X = Xs + X_D$$

$$= 1.98 \text{ inches}$$

The spring constant of this spring is:

$$K = \frac{d^4\ G}{64\ R^3\ N}$$

$$= \frac{(.059)^4\ 10.25 \times 10^6}{64\ (.11)^3\ 44}$$

$$= \frac{1210 \left(10.25\ (10\ )\right)}{2810\ \left[\ 1.33\ \right]}$$

$$K = 33.1 \ {}^{\text{lb.}}/\text{in.}$$

The maximum stress existing in this spring is:

$$S_s = \frac{16\ pr}{\pi\ d^3}\left[1 + \frac{.615}{C_1}\right]$$

$$= \frac{16\ (1.98 \times 33.1)\ \ .11}{\pi\ \ (.059^3)}\left[1 + \frac{.615\ (.059)}{2\ (.11)}\right]$$

$$S_s = 206{,}000 \text{ psi}$$

The minimum ultimate tensile strength of NS 355 wire is 380,000 psi. This corresponds to a yield shearing stress of 228,000 psi.

The minimum factor of safety on this basis for this spring is therefore:

$$F.S. = \frac{228{,}000}{206{,}000}$$

$$F.S. = 1.1$$

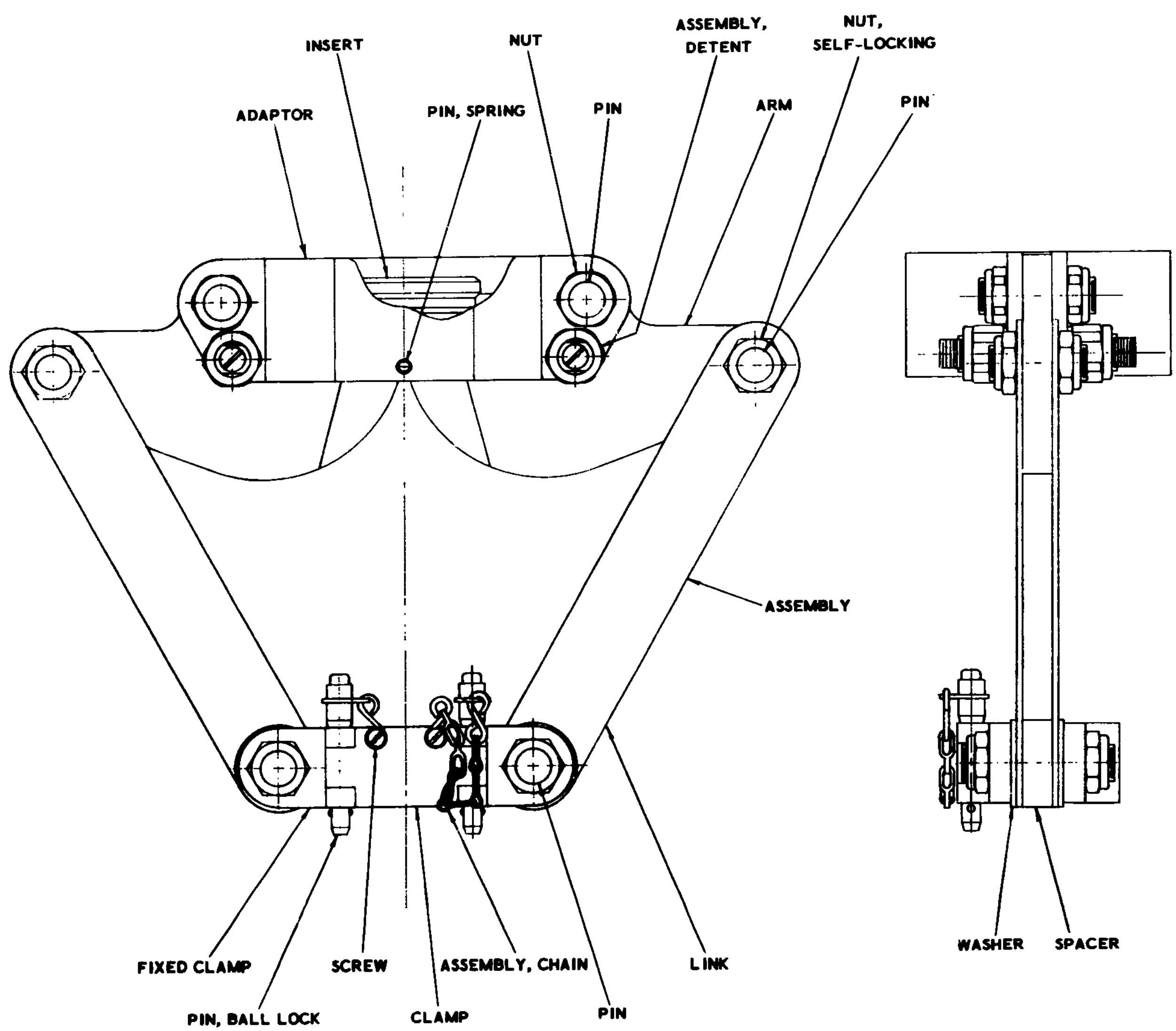
2. Retrieve Device.

a. Basic principles of operation

An adaptor which locks to the device by means of the locking keys in the same manner as the Large and Small Stake Adaptors is provided to withdraw emplaced stakes. The Adaptor consists of a link system which when actuated by a blow from the Device's Hammer translates the energy of the blow into an axial withdrawal force. An assembly drawing of this device is found on the following page.

b. Description and function of components

The Adaptor Assembly. This consists of two parts, the Adaptor and the Insert, which are screwed together to form the Adaptor Assembly. The Insert is essentially a short, thick-walled steel cylinder with a flange at its bottom end. This flange is threaded to enable the assembly of the Adaptor and Insert. A groove of the type used in the Stake Adaptors is furnished to rigidly attach the Assembly to the Device. The



Assembly Drawing of Retrieve Device

Figure 25

curved portion of the Arms pass through the hole in the Insert. The Adaptor is essentially a short, steel octagon with a large hole through its center. Threads are provided at the bottom of this hole to accommodate the Insert. Four ears, in two pairs, extend from opposite sides of the octagon. There are two holes in each ear. One of these is threaded. Each threaded hole accommodates a Detent Assembly which holds the Assembly in "Ready" position. The other hole accommodates a Pin which furnishes a pivot for the Arms. The Lock Ring of the Emplacement Device fits inside the large hole in the octagon. This enables the Key Lock System to engage the groove in the Insert.

<u>The Detent Assembly</u>. Each of the four Detent Assemblies associated with the Retrieve Device consists of the Body, Header, Retainer, Ball and Detent Spring. The Body is a hollow steel cylinder which is threaded externally throughout its length. One end is threaded internally for a short distance. This is to accommodate the external threads of the Retainer. The other end of the Body has an internal radius which enables it to retain the Ball when inserted. The Retainer is a short steel cylinder with external threads throughout its length. One end of the Retainer is slotted to enable it to be screwed into the Body. Its other end has a short cylindrical portion which fits inside the Detent Spring, positioning it. The Detent Spring is a helical compression spring which fits between the Retainer and Header. It pushes the Header against the Balls, pushing the Balls tight against the internal radius of the Body. (The Header is a short cylindrical steel piece which fits inside the Body between the Ball and the Spring.) While the Body of the Assembly is held fixed, the Ball can be pushed in, if a sufficiently large force is applied.

<u>The Arms</u>. The Arm is essentially a steel plate with two holes-- one at one end, and the other in the middle. One end of the Arm is thicker relative to the other and curves upward. The Arm is a short, high strength lever. It pivots about the Pin which passes through its center hole. This pin connects the Arm to the ears of the Adaptor Assembly.

A lock nut on each end of the Pin secures it. The thick, curved portion of the Arm passes through the hole in the Insert of the Adaptor Assembly. When the Arms are positioned before firing, the ends of the curved portion of the Arms are flush with the top of the Insert. They are each held in this position by two Detent Assemblies. The Balls of the Detent Assemblies engage dimples in the Arm located so as to accomplish this. Through the other end of the Arm is a hole which accommodates another Pin. This Pin furnishes a pivot for the Links and Washers.

<u>The Link</u>. Each of the four Links employed on the Device is a rectangle of steel sheet with a hole near each end. The Link connects the components mentioned above with the Clamp. Each Link is placed over its Pin and then a Washer is installed. A lock nut is then fastened to each end of the Pin.

<u>The Fixed Clamp</u>. This is a "U" shaped steel component with two ears protruding from each side. Through each pair of ears is a hole to join the two Links, Spacer, and Washers with each ear pair. Provision is also made on the Fixed Clamp to hinge the Clamp to it.

<u>The Clamp</u>. The Clamp is a rectangular steel piece with a circular portion removed. It has provisions for hinging onto the Fixed Clamp by means of the Hinge Pin. When the Clamp unit is closed, the Lock Pin retains it in this position. These pins are of the quick-release type and may be released by depressing the Detent on the Head of the pin.

<u>The GP-112/G Adaptor</u>. In the operation of the Retrieve Device while retrieving Ground Stake GP 113/G, the top end of the Clamp unit engages the bottom of the Stake head. To retrieve GP 112/G, an Adaptor is needed to adapt to this Stake. The Adaptor is a thick, hollow cylinder with internal and external shoulders. A hole is provided perpendicular to the centerline of the Adaptor to accommodate a quick-release pin. This pin is attached to the Adaptor and connects it and Stake GP 112/G. The

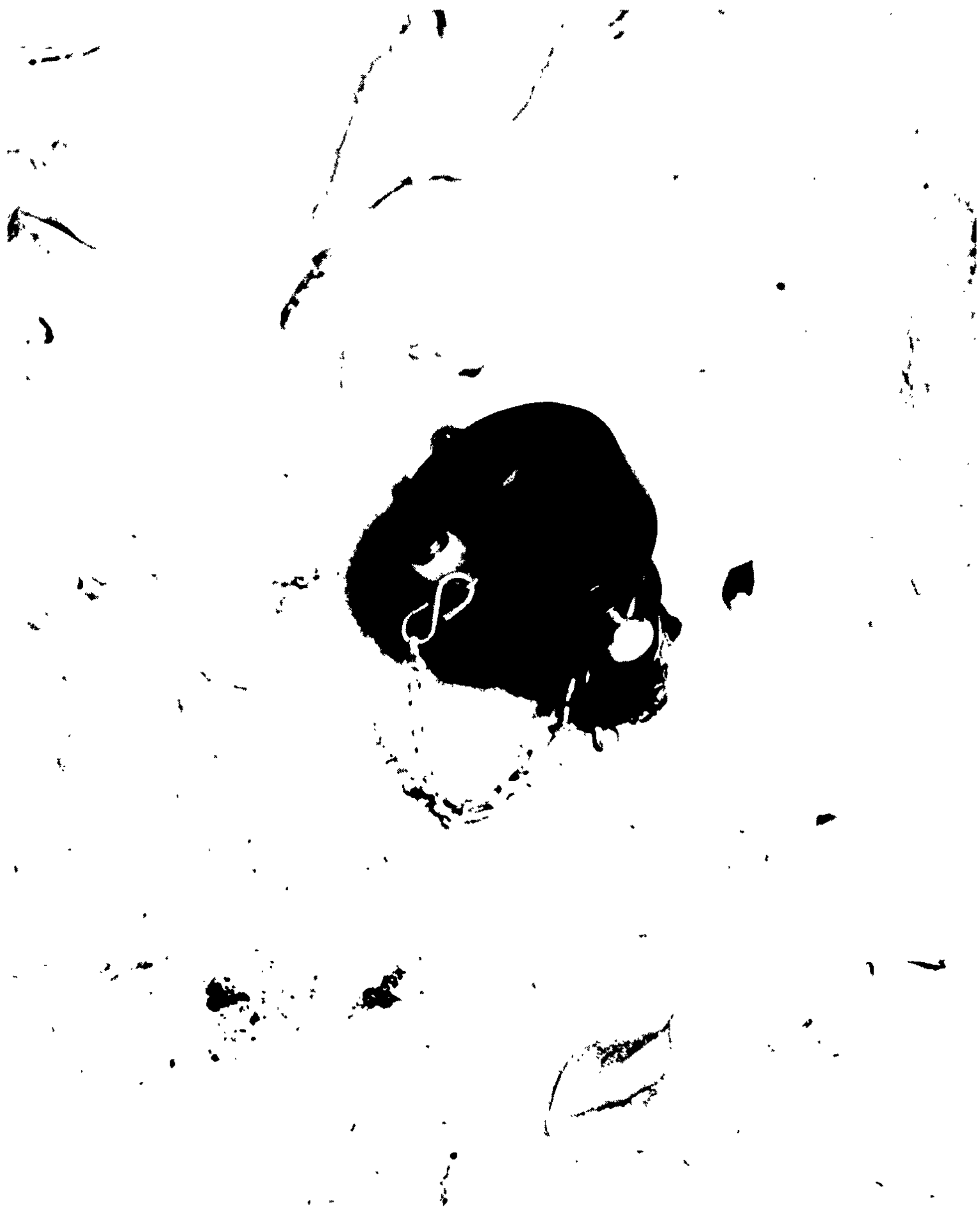
Adaptor fits inside the large hole in the Clamp unit. The external shoulder of the Adaptor enables the Retrieve Device to impart the retrieve force to it, while the pin transfers this force to the Stake. A picture of this Adaptor mounted on Stake GP 112/G is found on the following page.

(1) Operating Cycle

The preceding discussion was concerned with a physical and functional description of each component of the Retrieve Device. The interrelation of these in the overall device is explained in the following discussion. The Device is shown in position for the retrieval of Ground Stake GP 113/G in Figure 27 . In this position, the Detent Assemblies engage the dimples in the Arms to hold the Retrieve Device to this shape. The operator must lift the entire Device until the top of the Clamp unit engages the bottom of the head of the Stake. The Emplacement Device is rigidly attached to the Retrieve Device through the use of the Key System mentioned previously. Upon initiation of the Emplacement Device, the Hammer begins its downward travel. Instead of impacting a Stake, however, it impacts the curved ends of the Arms, which pushes them downward. This, in turn, moves the other end of the Arms upward. The Links subsequently pull on the Clamp unit and retrieval of the Stake begins. The upward load applied to the Stake by the Arm tends to pull the Emplacement Device and the Arm portion of the Retrieve Device downward. The inertia of these components, however, is sufficient to hold these parts essentially fixed in space during the time that the maximum upward load on the Stake acts.

To repeat the cycle of the Retrieve Device, the operator must pull it up until the Detent Assemblies again engage the dimples in the Arms. When this occurs, the Clamp unit will be engaging the bottom of the head of the Stake and the Emplacement Device can be initiated again, further retrieving the Stake.

The operating cycle is identical for the retrieval of Ground Stake GP 112/G except for the presence of the GP 112/G Adaptor. A picture of this Set-up is found on the following page.



GP-112/G Adaptor Installed on Stake

Figure 26

Retrieval of Ground Stake GP-113/G
Figure 27

Retrieval of Ground Stake GP-112/G

Figure 28

c. Stress Analysis

Because of the inherent complexity of the operating principle of the Retrieve Device, it was necessary to first initiate a kinematic design which was extremely conservative from a structural standpoint. After this was done, and the device performed satisfactorily, a strain gage program was initiated to measure the structural response of the device. A more efficient structure evolved from the results of this program.

The locations and directions of the highest stress in this Device are:

(1) Axial tension stresses in the Links
(2) Bending stresses in the filets of the Arms. This is the top curved portion of the Arm between the two Pins.
(3) Tension Stresses in the top of the rim of the Adaptor Assembly.

These stresses have the values shown below:

(1) Links . . . . 35,600 psi
(2) Fillets . . . . 88,800 psi
(3) Rim of Adaptor Assembly.. 19,600 psi

This provides a desirable factor of safety throughout the Device. The minimum factor of safety, based on the yield point of the steel used, is:

$$F.S. = \frac{.6\ (185,000}{88,800}$$

$$F.S. = 1.25$$

3. Emplacement and Retrieve Device

a. Materials Selection

The materials selected for the major components of

the entire Ballistic Hammer are found in the table following. Where applicable, the selection of materials was based on the results of the stress analysis, and the life cycle calculations. It should be pointed out that all materials selected are readily available and that as many fabricated metal parts, as was practicable, are of corrosion resistant steel.

(1) Emplacement Device

PART NAME	MATERIAL
Barrel	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
Head	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
Brace	Type 304 corrosion resistant steel.
Buffer	Butyl rubber, 80-90 Durometer.
Hammer	4130 Steel, heat treated to a tensile strength of 185,000 psi.
Exhaust Manifolds	Type 304 corrosion resistant steel.
Exhaust Pipe-Handles	Type 304 corrosion resistant steel tubing.
Trigger	Type 17-7 PH condition "A" corrosion resistant steel, heat treated to a tensile strength of 220,000 psi.
Large Stake Adaptor	Type 7075 Aluminum, having a tensile strength of 65,000 psi.
Small Stake Adaptor	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
Hammer Return Spring	Type NS 355 corrosion resistant steel spring wire.
Bracket	4130 Steel, heat treated to an ultimate strength of 185,000 psi.

PART NAME	MATERIAL
Link	4340 Steel, heat treated to an ultimate strength of 185,000 psi.
Spring Guide	4130 Steel, heat treated to an ultimate strength of 185,000 psi.
Magazine Support	4340 Steel in the annealed condition.

(2) Retrieve Device ---- All major structural components of the Retrieve Device are fabricated from 4130 or 4340 steel, heat treated to an ultimate strength of 185,000 psi.

As stated previously, as many components as was possible from a structural standpoint were fabricated from corrosion resisting steels. After heat treating, these corrosion resisting steel parts were descaled completely in sodium hydride caustic soda, and passivated per MIL STD 171A, finish No. 5.4.2. After this, all of the standard steel and corrosion resisting steel parts (with the exception of the Type 304 corrosion resisting steel parts) are finished per MIL P 16232B Type M, Class 2. This is a phosphate coating identical to that applied to the caliber 30 action. The Type 304 corrosion resisting steel parts are sand-blasted after all fabrication, which produces their particular surface finish.

The factors which influenced the material and finish selections are listed below and defined. Aside from corrosion caused by the direct attack of chemicals on the Device, some erosive corrosion is present when the exhaust gases are made to change direction for porting. However, since the exhaust velocities and gas temperatures are relatively low, and only a minimum of solid materials is present. This problem is not severe in this application.

Crevice Corrosion

Often when subjected to liquid corrosive agents, accumulation occurs in crevices where destructive action continues after exposed areas dry. The basic action occurs when there is a difference in oxygen concentration or metal ion concentration causing the material to flow

from a positive and negative electrolyte giving rise to electrochemical action. Obviously, the remedy was to reduce and avoid unnecessary crevices wherever possible.

Dezincification

This corrosive phenomenon occurs with copper-zinc alloys resulting in the loss of zinc from the alloy. Both layer and plug type dezincification occur leaving a porous and weak surface.

Since no copper alloys are used, this is no problem area.

Direct Attack

Direct attack by corrosive media was prevented by the proper selection of materials. In addition, oxidizing agents were used to form surface oxides or layers of absorbed oxygen which increase the resistance of base materials to chemical attack. Such a process, passivation was used on all stainless steel parts.

Stress Corrosion

When stressed metals are exposed to a corrosive environment deterioration occurs resulting in the form of localized cracks. This promotes material failure at lower stress levels than those at which the material ordinarily yields. Each metal is susceptible to different agents in different degrees. The proper selection of materials is the best precaution after deciding the operating environment.

b. Weight & Size Analyses

The weights of the major components of Ballistic Hammer MX-6321 ( )/G are tabulated below. These weights have been adjudged as the optimum for each component, based on the results of the analytical and experimental stress analyses conducted in this area.

	Item	Weight (lbs)
(1)	Basic Hammer with no Magazine or Adaptors attached	31.5
(2)	Ground Stake GP-113/G Emplacement Adaptor	.85
(3)	Ground Stake GP-112/G Emplacement Adaptor	1.77
(4)	Basic Retrieve Device	17.00
(5)	Retrieve Adaptor for Ground Stake GP-112/G	1.43
(6)	Eighteen (18) Filled Magazines (Total 270 rounds)	7.00
(7)	Tool Roll	7.00
	Total	66.55

Other pertinent weight and overall lengths of the system are found below:

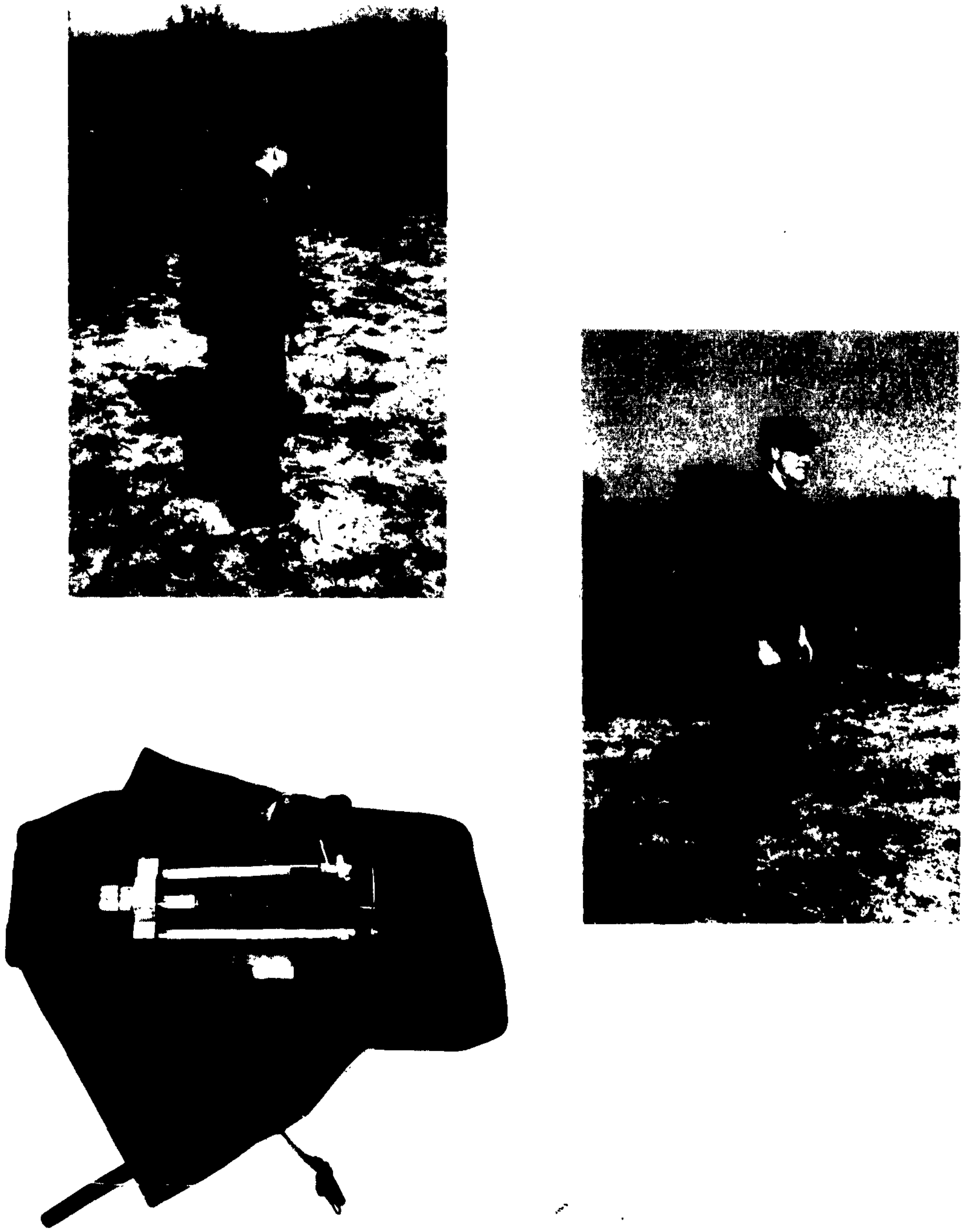
	Operation	Weight (lb)	Overall Length (in.)
1.	Emplacement Mode for Ground Stake GP-113/G*	32.74	30.6
2.	Emplacement Mode for Ground Stake GP-112/G*	33.66	27.5
3.	Retrieve Mode for Ground Stake GP-113/G* (See Fig. 27)	48.89	32.1
4.	Retrieve Mode for Stake GP-112/G* (See Fig. 28)	50.32	32.1

* Includes weight of one filled 15-round magazine.

4. Tool Roll

A tool roll has been furnished in order to facilitate the manual transportation of Ballistic Hammer MX-6321 ( )/G in the field. The tool roll has provisions for all components used in the emplacement and retrieve of both Ground Stakes.

A picture of the tool roll with the flaps open is found on the following page. As can be seen from this picture, each individual item is readily accessible, which is a very desirable feature due to the variety of the tasks which will be undertaken by the tool. The two pictures of the Tool Roll mounted on the operator's back show the mounting arrangements are similar to those of the standard full Field Pack with the addition of two straps which attach to the operator's belt area. These transfer a portion of the total weight from his back to his waist. (The harness portion of the roll was designed according to MIL-P-3392).



Breadboard Model of Ballistic Hammer

Figure 29

D. Testing Program

This portion of the report describes the various tests which were performed to evaluate the performance of the Device. During the early phases of the program tests were conducted to measure energy required to emplace Stakes GP-112/G and GP-113/G into representative soils and permafrost. In order to obtain ballistic test data, a breadboard device (shown in Figure 30) was designed and tested to obtain further emplacement and some limited retrieve data. The test procedures and results obtained were described in detail in the Design Plan and will not be repeated in this report as their applicability to the engineering model is limited.

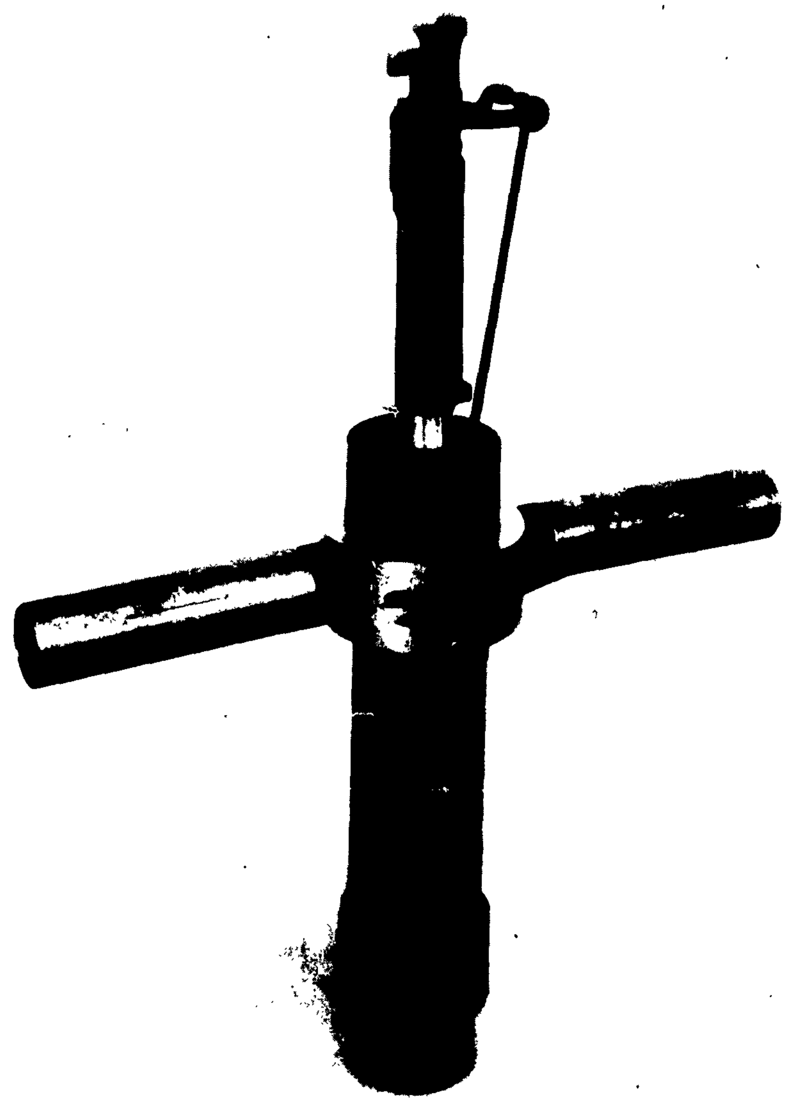
Tests performed with the engineering model were employed as an aid to development and to evaluate performance of the device after it was built. Tests of the first type included:

Hammer velocity measurements
Tool recoil velocity measurements
Bolt velocity measurements

Tests of the second type included:

No load firing
Locked shut firing
Human Factors
Flash & Smoke
Noise
Holding Power
Emplacement-Retrieve Capability
Stake Life
Strain Gage Evaluation

In addition to these tests, a third series was authorized near the conclusion of the program to evaluate the effect of hammer momentum upon stake driving efficiency.



Breadboard Model of Ballistic Hammer

Figure 30

1. No-load and Locked-shut Firings

These tests were conducted to determine the extent of the effects, if any, on the Device if it were subjected to extremely severe firing conditions.

a. No-load Firing

In this test, the Head of the Device was rigidly mounted to a firing table and the Device was initiated as in regular operation. The bolt of the action was allowed to cycle. The Hammer was free to travel and, since no stake was in place, impacted the Buffer System of the Barrel. The stress in the Barrel from this was measured by strain gages and was found to be 68000 psi. This is considerably below the yield point of the Barrel material. A detailed inspection revealed that there was no damage to any components of the Device. This data was collected in a series of five tests. In addition to these planned tests there were several instances during functional testing when the tool was not positioned correctly before firing and the no-load situation was produced. No adverse effect upon either operator or Device was noted as result of these firings.

b. Locked-shut Firings

These consisted of three tests - in all tests the head of the Device was rigidly mounted to a firing table:

(1) The bolt of the action was restrained from movement, but the Hammer was free to travel and impact the Buffer System.

(2) The bolt of the action was allowed to cycle, but the Hammer was restrained from movement.

(3) The bolt of the action and the Hammer were restrained from any movement.

The Device was fired under each of the three conditions above. After each test, every component of the Device was thoroughly inspected. This revealed that no component sustained any damage under any of the above firings. This verified the analytical determination of the capability of the device with respect to these conditions.

It should be pointed out that the stress in the Barrel for Case "b-(1)" is identical to that for Case "a". The maximum pressure encountered in the head for Cases "b-(2)" would be 20,600 psi.

After the conclusion of the above tests, the Device was found to function normally in all phases of operation.

2. Human Factors

a. Reaction to Recoil

During the course of development the determination of the degree of recoil tolerable to the operator without fatigue was conducted. To ensure that representative findings were obtained, the reactions of several operators were evaluated. These tests led to the establishment of the present configuration, which incorporates a 4-pound Hammer moving at a velocity of 116 ft/sec. This corresponds to recoil impulse and velocity of:

$$I = \frac{4}{32}(116)$$

$$= 14.5 \text{ lb-sec}$$

$$V = \frac{I}{M}$$

$$V = 15 \text{ ft/sec}$$

This has been established as the maximum velocity-impulse combination which the operator can withstand over prolonged periods of exposure. The recoil distance encountered with this impulse in the Device varies from 5 inches with Ground Stake GP-113/G to 4 inches with Ground Stake GP-112/G. (The effects of recoil impulse and distance are negligible during the retrieval operation.) At one stage in the development, over 500 rounds were fired by one operator within a 6 hour period. Approximately 90% of the rounds were for emplacement, with the remaining 10% for retrieval. Three-fourths of the total rounds were expended on Ground Stake GP-113/G. At the conclusion of this, absolutely no adverse effects of any type were observed by the operator, either immediately after exposure, nor for prolonged periods after exposure.

b. Exhaust Gases

As was previously mentioned, the propellant gases push on the small end of the Hammer for a Hammer travel of one inch. They are then exhausted through the Head and eventually exit through the Mufflers. These gases possess considerable pressure and velocity, even at the point of exit from the Mufflers. In addition to this, they could contain small particles of unburned propellant or propellant ash, traveling at high velocities. For these reasons, it is recommended that personnel come no closer than a distance of six feet from the exit plane of the Mufflers during operation of the Device. Personnel working constantly with the Device during its developmental stages experienced no adverse or uncomfortable effects from these gases.

c. Flash and Smoke

(1) Flash

There is no flash associated with the Device in any phase of operation, including conditions of reduced visibility. This is because the propellant gases are completely burned in the high pressure chamber of the Head before exhasustion. They are then expanded throughout a long distance and exhausted out to the atmosphere, at which stage they are "cold", relative to their original state. Observation of the Device under dusk light conditions revealed no flash.

(2) Smoke

A small amount of grey-black smoke is emitted from the Mufflers with each shot. The quantity and color of this smoke is such as to be considered negligible with respect to any effects on operating conditions or ability to detect the operation or location of the Device.

The smoke is non-irritating and is of such low quantity that it is quickly difused into the air upon exhaustion, making it difficult for the operator to detect by odor. Observation of the Device from a measured distance showed the smoke to be invisable at approximately 200 ft. in daylight.

d. Control Considerations

The ability of the operator to manipulate the controls and perform the duties required to normally operate the tool were evaluated with the operator clothed in normal clothing and in heavy winter clothing. There are four critical duties which concern the operator with respect to the clothing consideration:

(1) Ability to pull upon the Lock Ring to install pertinent Adaptors.

(2) Ability to actuate the Trigger

(3) Ability to exert the preloads upon the Handles.

(4) Ability to insert and pull out magazines.

The sizes and shapes of the controls pertinent to each of these operations have been arranged so that the operator will have no difficulty in performing these duties with either normal or heavy winter clothing on. This has been verified during the testing programs undertaken.

e. Noise

An important consideration with the Ballistic Hammer is its aural signature. During the course of this program it was found that the highest noise intensity was encountered during the emplacement of Ground Stake GP-113/G. This noise arises from the metal to metal impact of the Hammer and Stake, with ballistic noise contributing a negligible amount to overall noise intensity.

It was reasoned that the most meaningful evaluation of the noise produced could be made by stationing observers at various distances from the Ballistic Hammer during the emplacement of Ground Stake GP-113/G and evaluating their reactions. Two observers were used - their reactions are noted below:

Distance From Tool (ft.)	Noise Observations
400	Audiable
500	Still audiable, but of low intensity
600	Nearly faded out
700	Virtually silent

It was found that the prevailing weather conditions (clear or cloud cover) had no effect on the above observations.

3. Holding Power Tests

The immediate objective of these tests was to evaluate any difference in holding power between stakes emplaced (under identical soil and loading conditions) by the Ballistic Hammer, and those emplaced by an 8 pound sledge. The concept of these tests was to emplace the stakes in near proximity of each other under identical soil conditions by the two methods. A load was then applied perpendicular to the longitudinal axis of the stake through a 1-1/2 ton capacity chain fall. This load was measured by a 10,000 pound capacity dynamometer. The deflection of the stake corresponding to each particular load was measured by the relative motion of a pointer attached to the stake and a stationary scale. The results of these tests can be found on the following pages.

These results indicate that, for any particular soil conditions, there was little effect on the holding power of the stakes at any stage of deflection induced by the emplacement method.

A complete summary of the data recorded is found in the table following the test results.

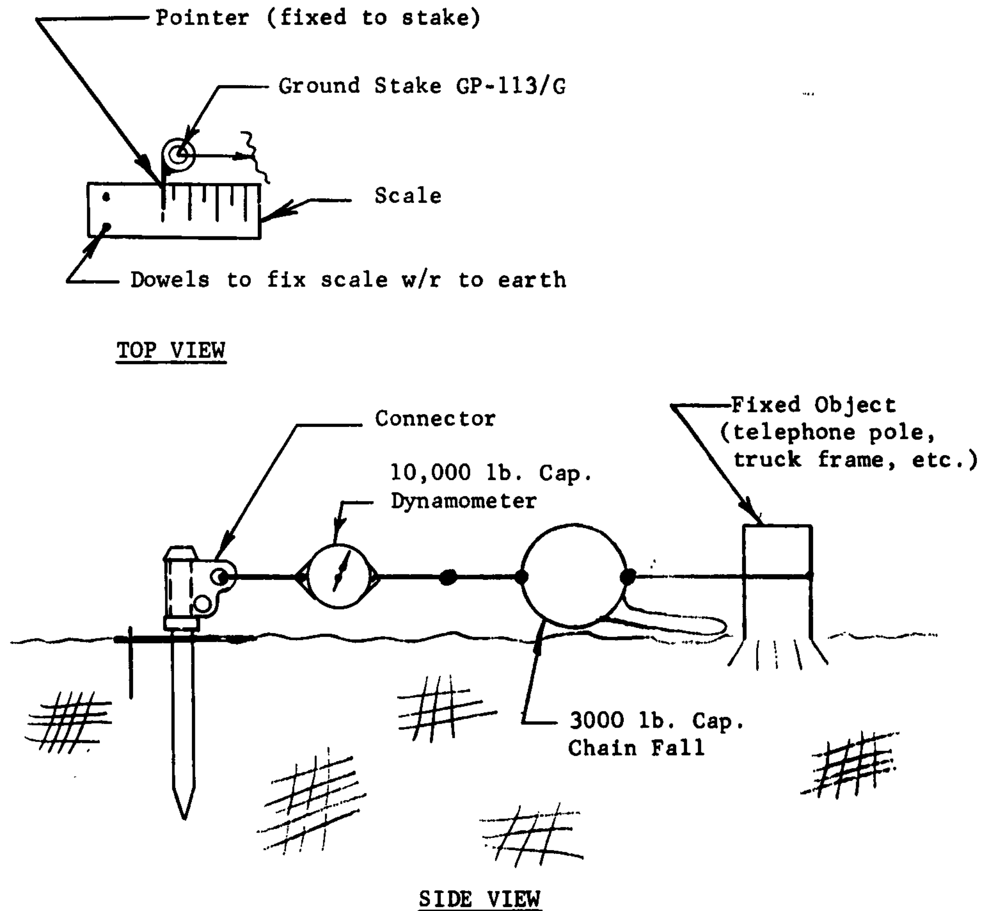
Figures 32 through 42 illustrate the plotted results of these tests. Government representatives observed the test set-up and a representative group of these tests.

4. Emplacement Capability

During the development of the Device, emplacement tests were conducted with Hammer energy and momentum as varying parameters. Figure 30 illustrates the breadboard device employed to obtain preliminary data.

The data presented below has been established for the Device in the "as-delivered" condition - a 4 pound Hammer with an impact velocity of 116 ft/sec. and a recoiling weight of 32.74 pounds*. The data presented serves as an indication of the emplacement capability of the Device and represents complete emplacement. For each particular situation, except for Group "1-(b)",

* For Ground Stake GP-112/G, this weight is 33.66 pounds.



SCHEMATIC OF PULL TEST APPARATUS SETUP
(Typical for Ground Stake GP-112/G)

Figure 31

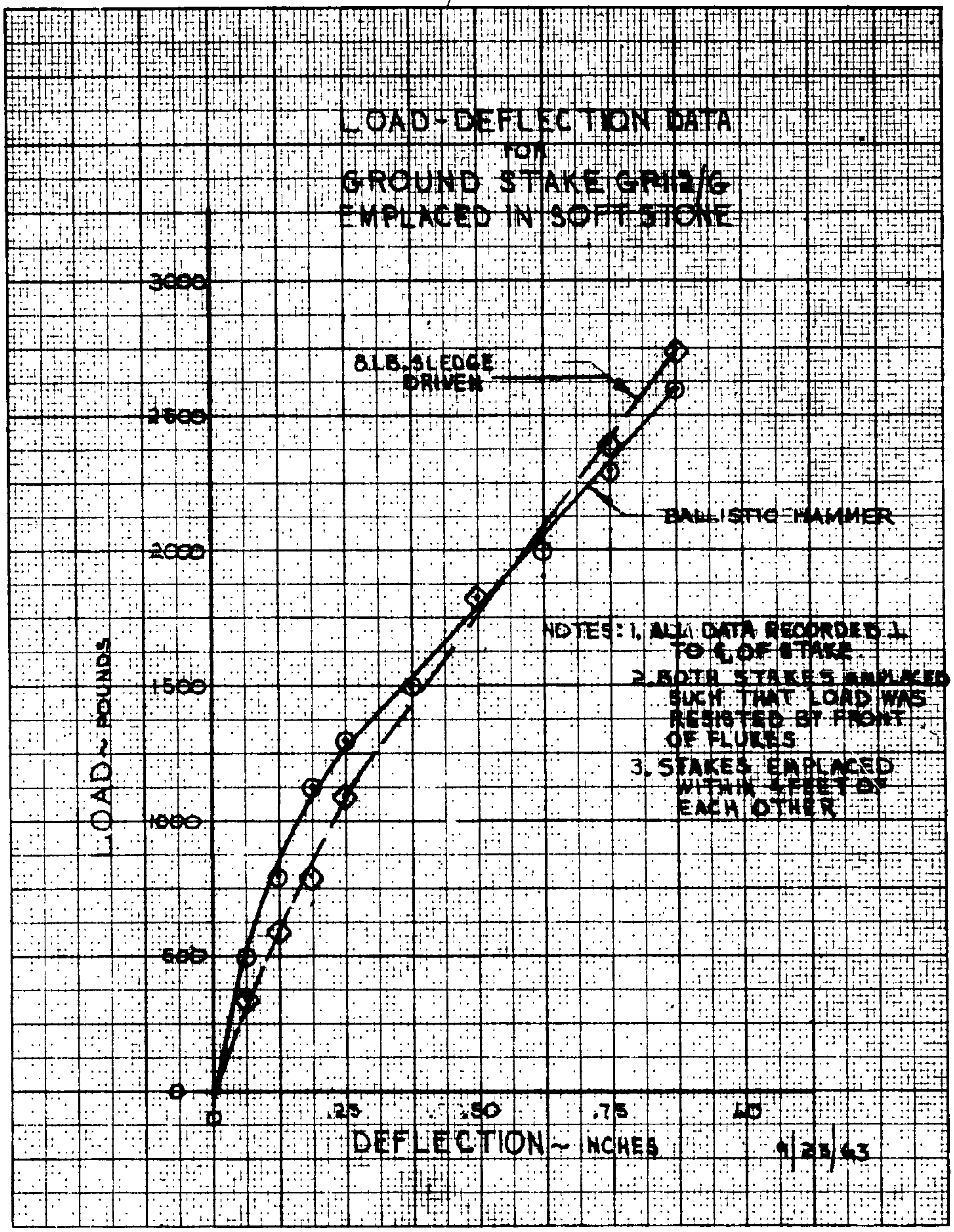

LOAD-DEFLECTION DATA
FOR
GROUND STAKE GR12/G
EMPLACED IN SOFT STONE
8 LB. SLEDGE DRIVEN
BALLISTIC HAMMER
NOTES: 1. ALL DATA RECORDED ⊥ TO ℄ OF STAKE
2. BOTH STAKES EMPLACED SUCH THAT LOAD WAS RESISTED BY FRONT OF FLUKES
3. STAKES EMPLACED WITHIN 4 FEET OF EACH OTHER
3000
2500
2000
1500
1000
500
0
LOAD ~ POUNDS
0
.25
.50
.75
1.0
DEFLECTION ~ INCHES
9/25/63

Figure 32

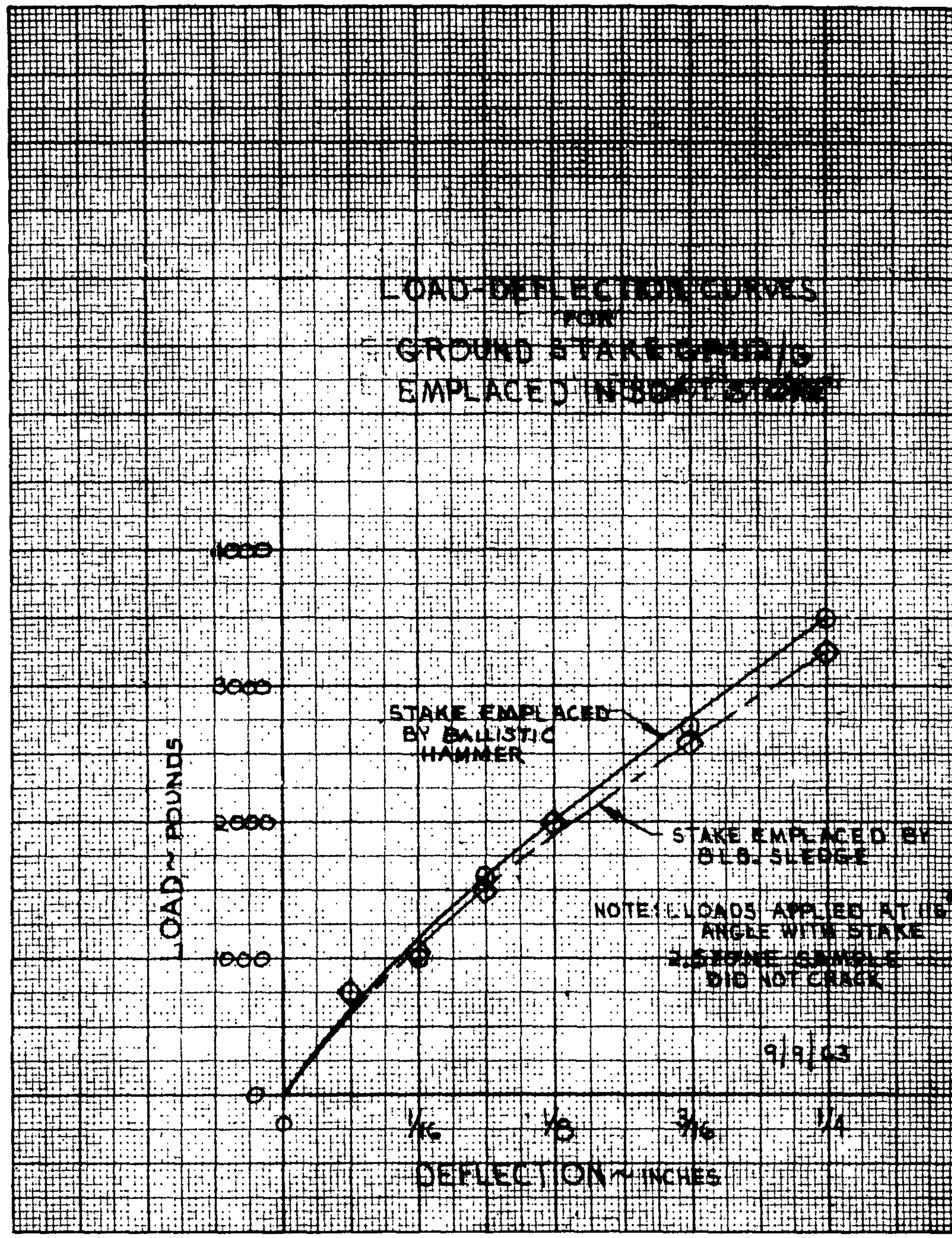
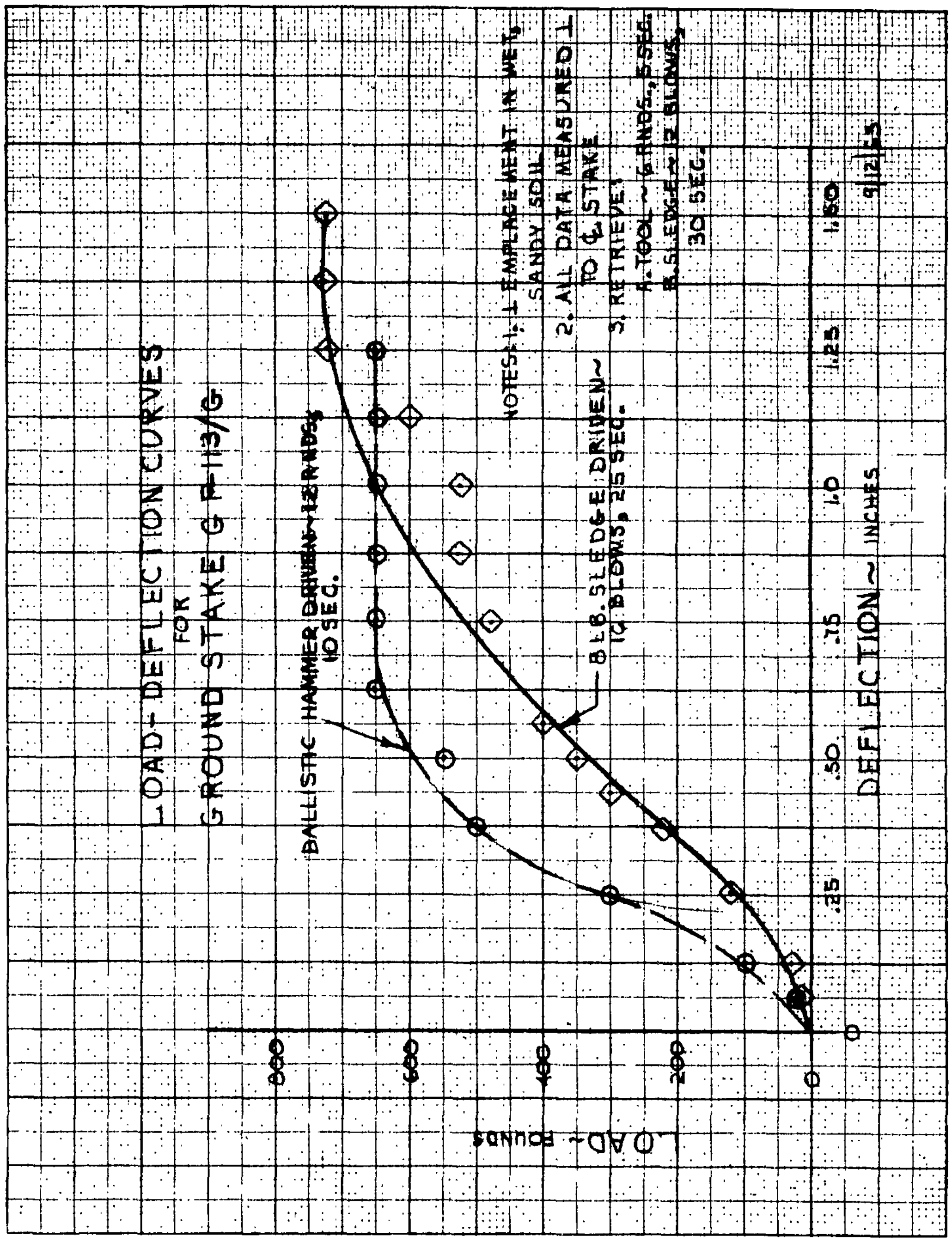

LOAD-DEFLECTION CURVES
FOR
EMPLACED
STAKE EMPLACED BY BALLISTIC HAMMER
STAKE EMPLACED BY 8LB. SLEDGE
NOTE: LOADS APPLIED AT
ANGLE WITH STAKE
DID NOT CRACK
9/9/63
LOAD ~ POUNDS
4000
3000
2000
1000
0
1/16
1/8
3/16
1/4
DEFLECTION ~ INCHES

Figure 33



LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP-113/G
BALLISTIC HAMMER DRIVEN~12 RNDS, 10 SEC.
8 LB. SLEDGE DRIVEN~ 10 BLOWS, 25 SEC.
NOTES: 1. EMPLACEMENT IN WET, SANDY SOIL
2. ALL DATA MEASURED ⊥ TO ℄ STAKE
3. RETRIEVE:
A. TOOL ~ 6 RNDS., 5 SEC.
B. SLEDGE ~ 12 BLOWS, 30 SEC.
LOAD ~ POUNDS
0
200
400
600
800
DEFLECTION ~ INCHES
0
.25
.50
.75
1.0
1.25
1.50

Figure 34

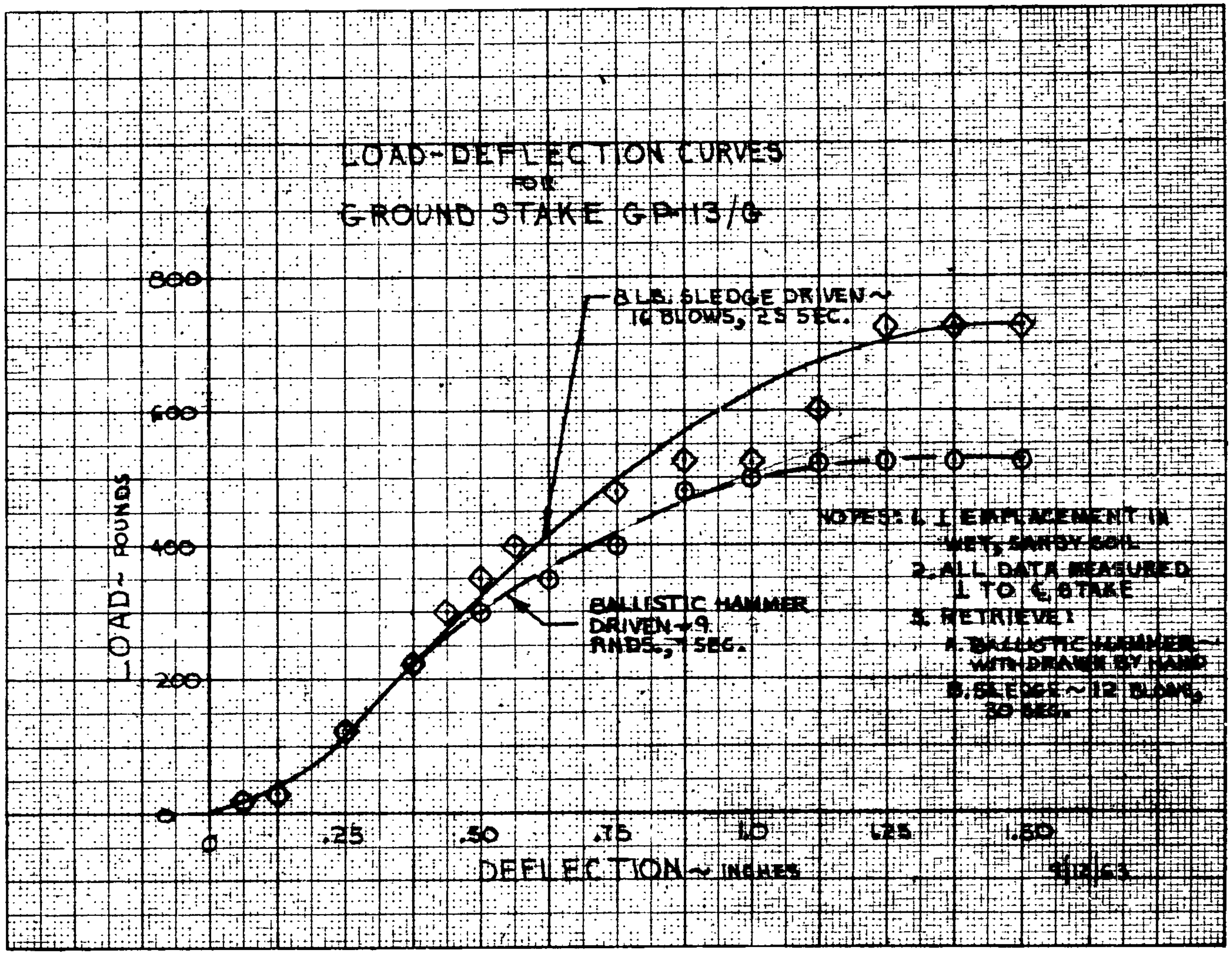

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP-13/G
8 LB. SLEDGE DRIVEN ~
16 BLOWS, 25 SEC.
BALLISTIC HAMMER
DRIVEN ~ 9
RNDS., 7 SEC.
NOTES: 1. EMPLACEMENT IN
WET, SANDY SOIL
2. ALL DATA MEASURED
⊥ TO ℄ STAKE
3. RETRIEVE
A. BALLISTIC HAMMER ~
WITHDRAWN BY HAND
B. SLEDGE ~ 12 BLOWS,
30 SEC.
LOAD ~ POUNDS
0
200
400
600
800
0
.25
.50
.75
1.0
1.25
1.50
DEFLECTION ~ INCHES

Figure 35

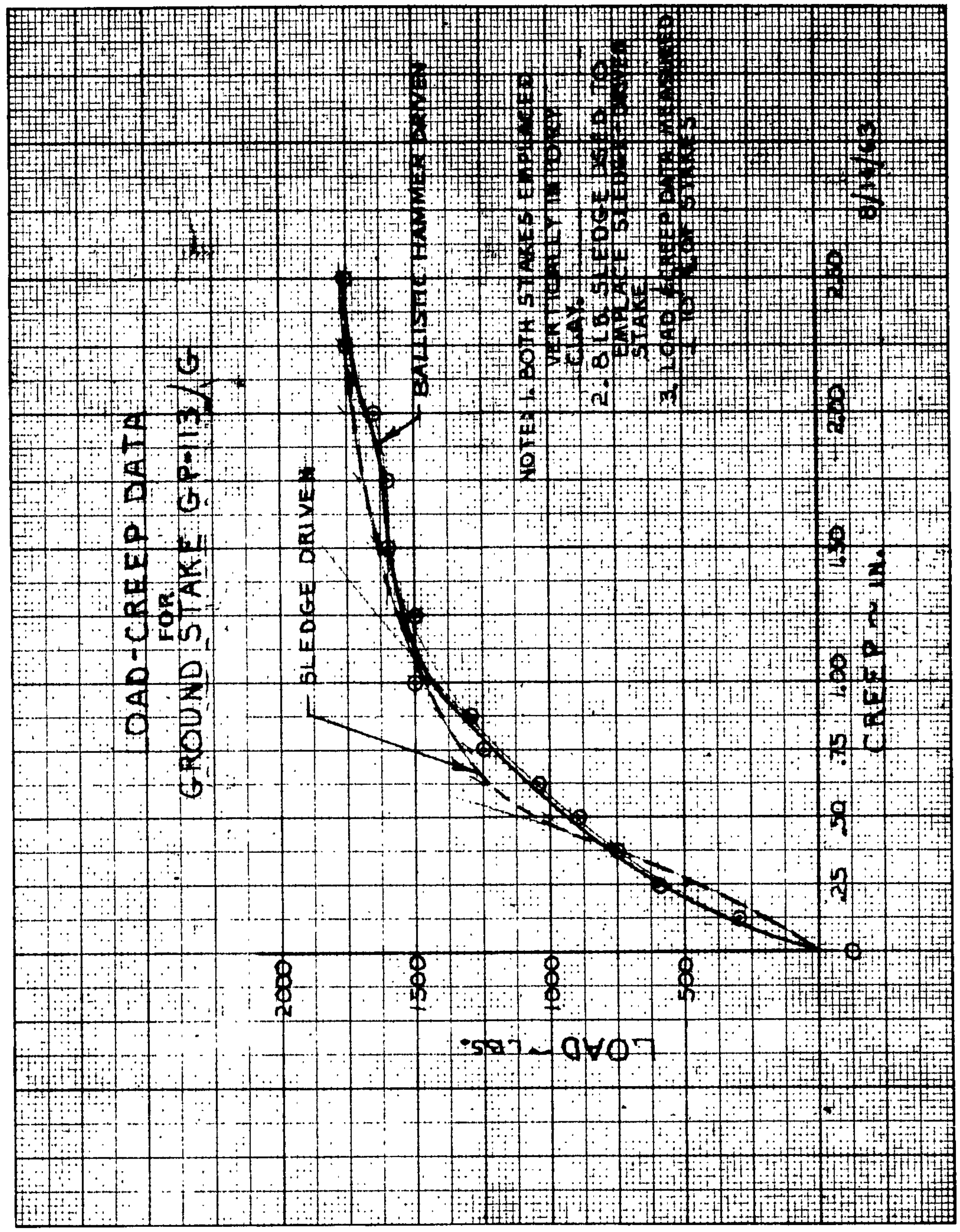

OAD-CREEP DATA
FOR
GROUND STAKE GP-13/G
SLEDGE DRIVEN
BALLISTIC HAMMER DRIVEN
NOTE: 1. BOTH STAKES EMPLACED VERTICALLY IN DRY CLAY.
2. 8 LB. SLEDGE USED TO EMPLACE SLEDGE DRIVEN STAKE
8/14/63
LOAD ~ LBS.
CREEP ~ IN.
500
1000
1500
2000
0
.25
.50
.75
1.00
1.50
2.00
2.50

Figure 36

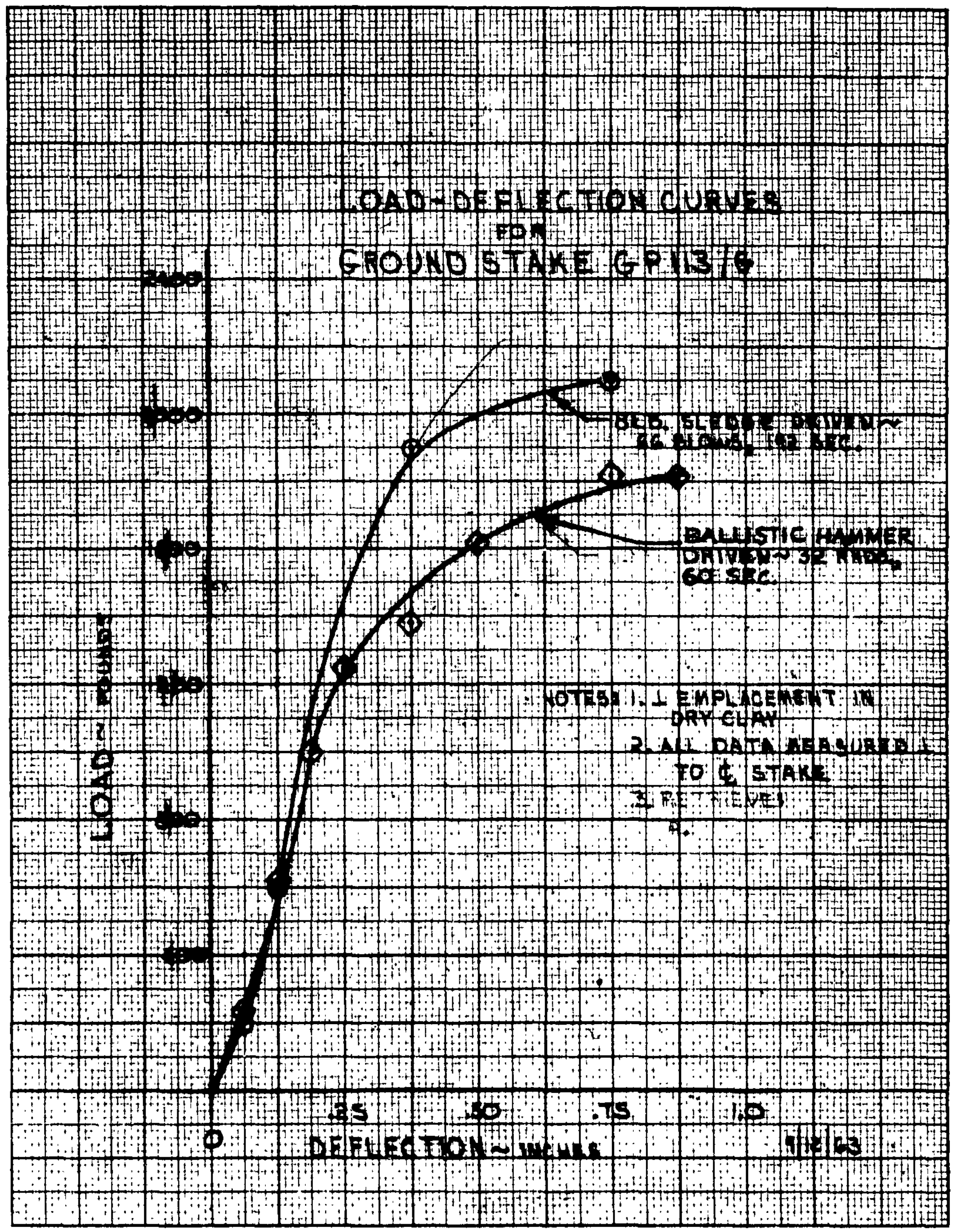

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP113/G
2400
BALLISTIC HAMMER
DRIVEN ~ 32 RNDS,
60 SEC.
NOTES: 1. ⊥ EMPLACEMENT IN
DRY CLAY
2. ALL DATA MEASURED ⊥
TO ℄ STAKE
LOAD ~ POUNDS
0
.25
.50
.75
1.0
DEFLECTION ~ INCHES

Figure 37

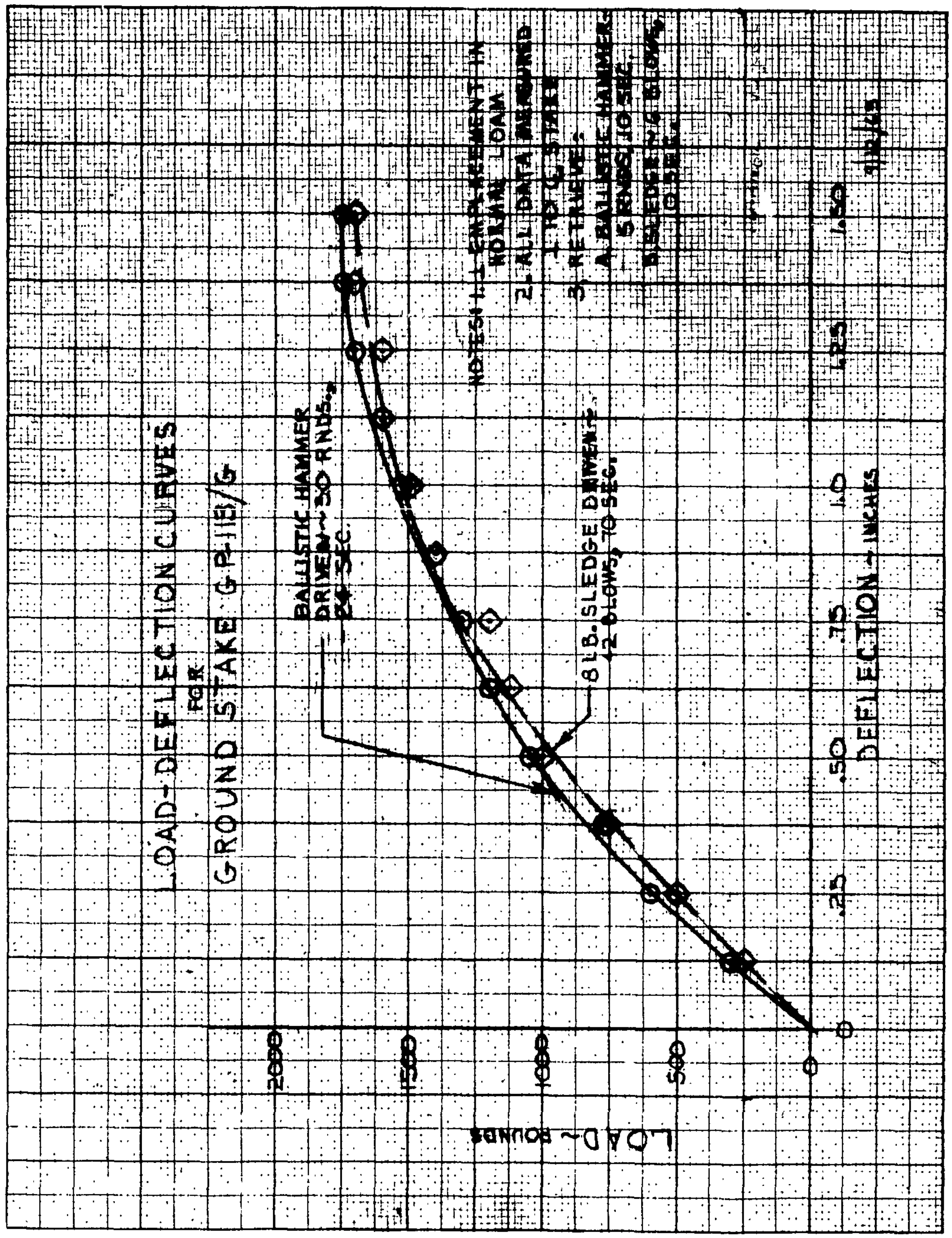

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE G.P.-118/G
BALLISTIC HAMMER
24 SEC.
8LB. SLEDGE DRIVEN
2000
1500
1000
500
0
LOAD ~ POUNDS
0
.25
.50
.75
1.0
1.25
1.50
DEFLECTION ~ INCHES

Figure 38

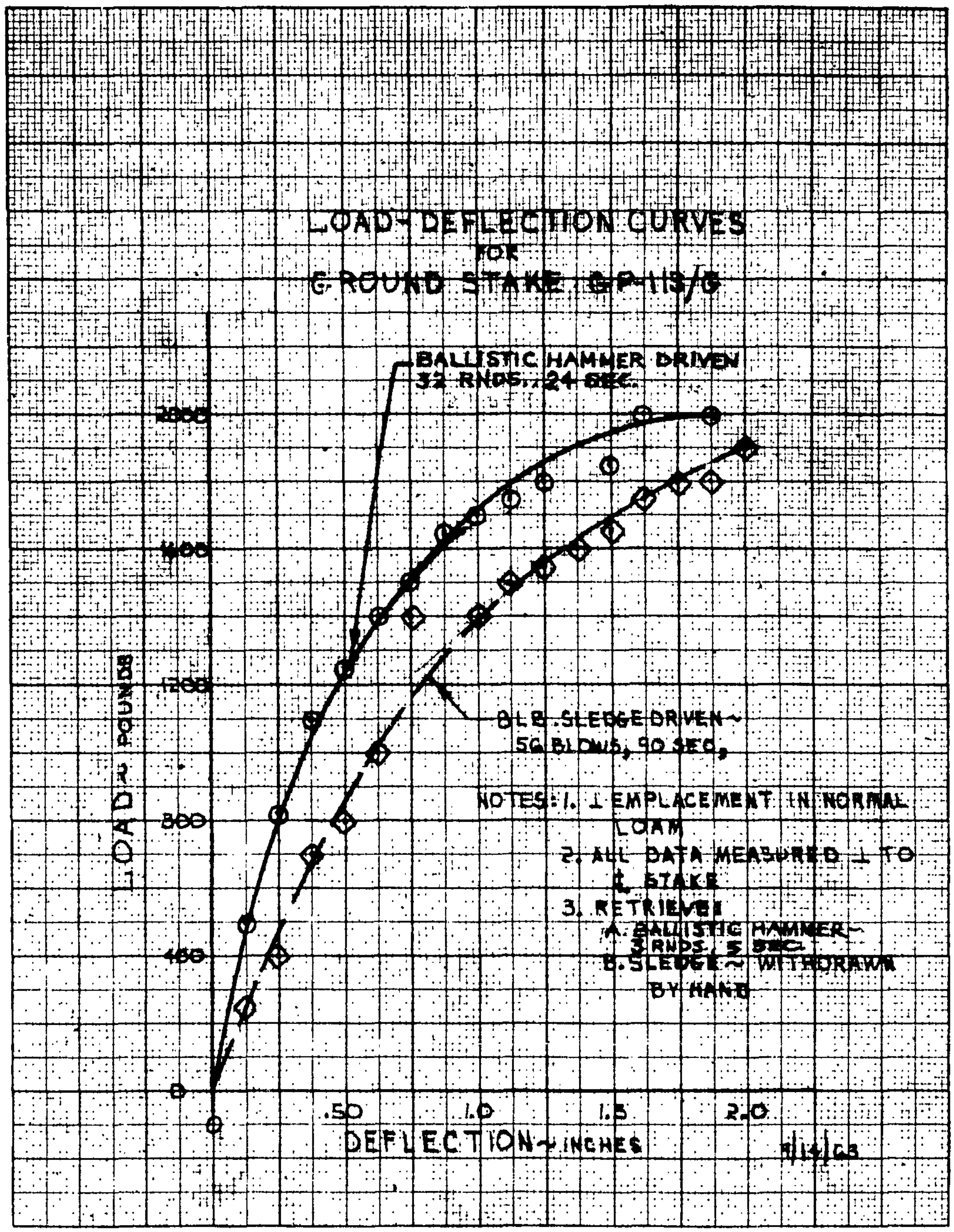

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP-113/G
BALLISTIC HAMMER DRIVEN
32 RNDS., 24 SEC.
8 LB. SLEDGE DRIVEN ~
56 BLOWS, 90 SEC.
NOTES: 1. ⊥ EMPLACEMENT IN NORMAL LOAM
2. ALL DATA MEASURED ⊥ TO ℄ STAKE
3. RETRIEVE:
A. BALLISTIC HAMMER ~ 3 RNDS, 5 SEC.
B. SLEDGE ~ WITHDRAWN BY HAND
LOAD ~ POUNDS
DEFLECTION ~ INCHES
2000
1600
1200
800
400
0
.50
1.0
1.5
2.0

Figure 39

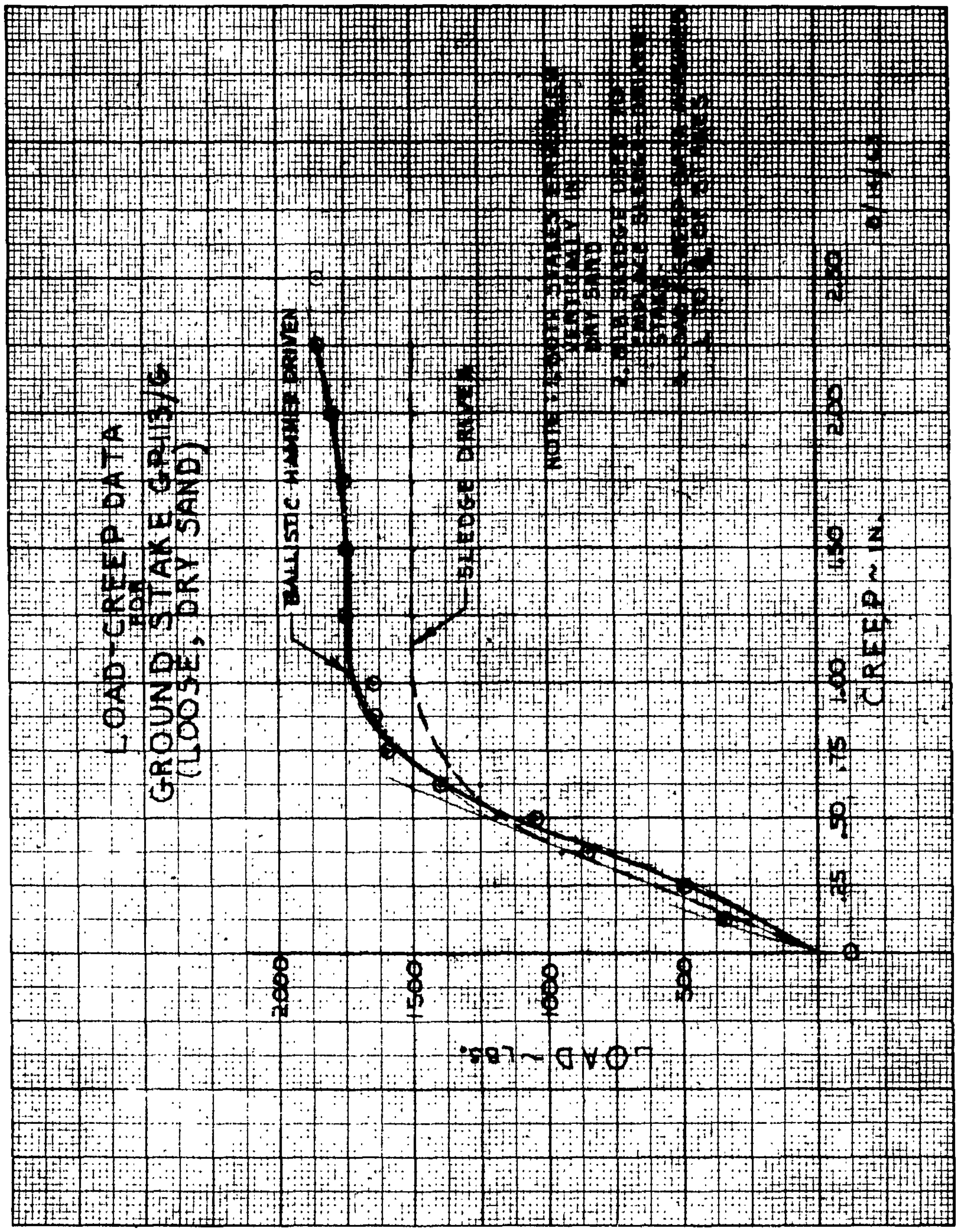

LOAD-CREEP DATA
FOR
GROUND STAKE GR-15/6
(LOOSE, DRY SAND)
BALLISTIC HAMMER DRIVEN
SLEDGE DRIVEN
LOAD ~ LBS.
2000
1500
1000
500
CREEP ~ IN.
.25
.50
.75
1.00
1.50
2.00
2.50

Figure 40

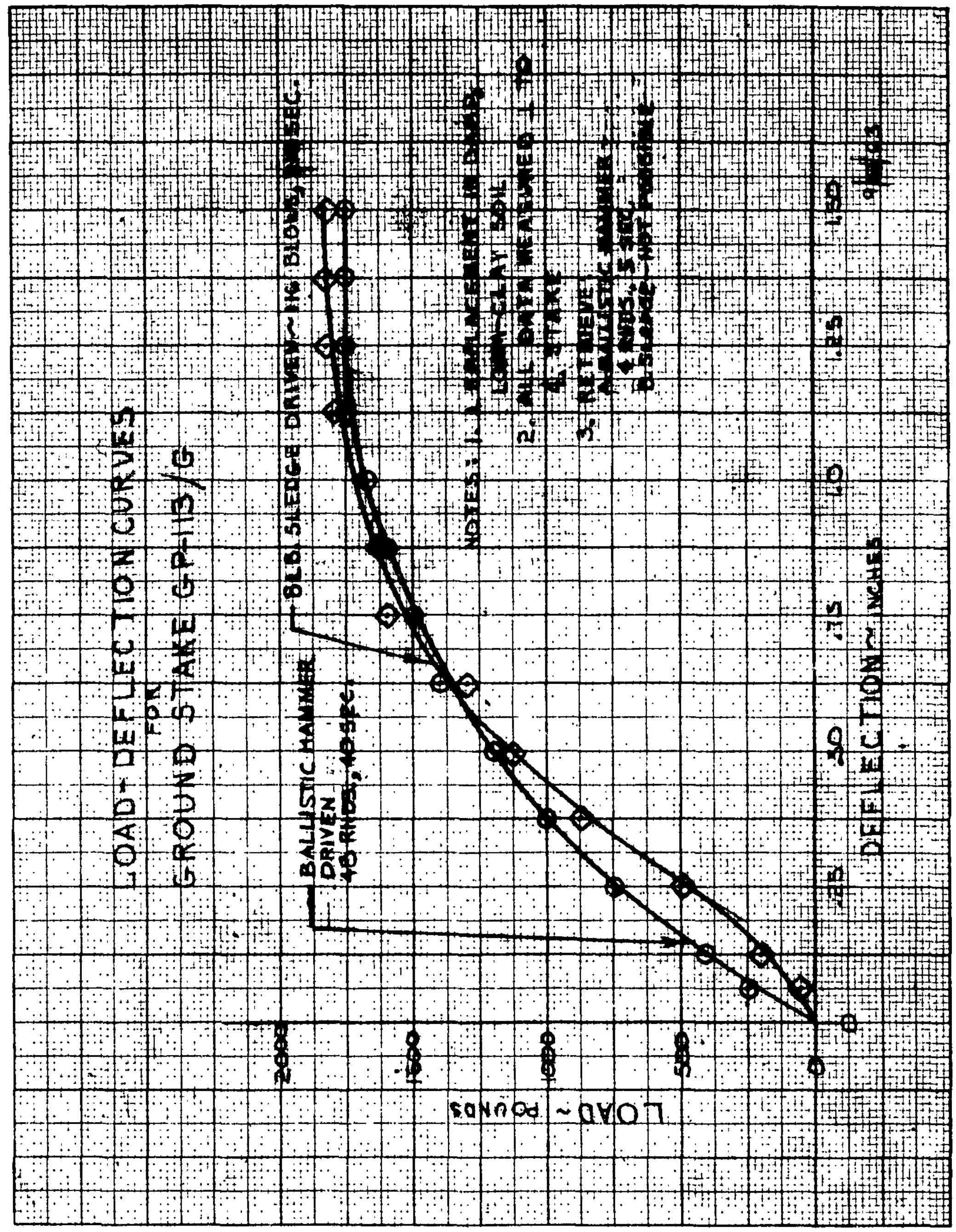

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP-113/G
BALLISTIC HAMMER DRIVEN
8 LB. SLEDGE DRIVEN
NOTES:
LOAD ~ POUNDS
DEFLECTION ~ INCHES
0
500
1000
1500
2000
.25
.50
.75
1.0
1.25
1.50

Figure 41

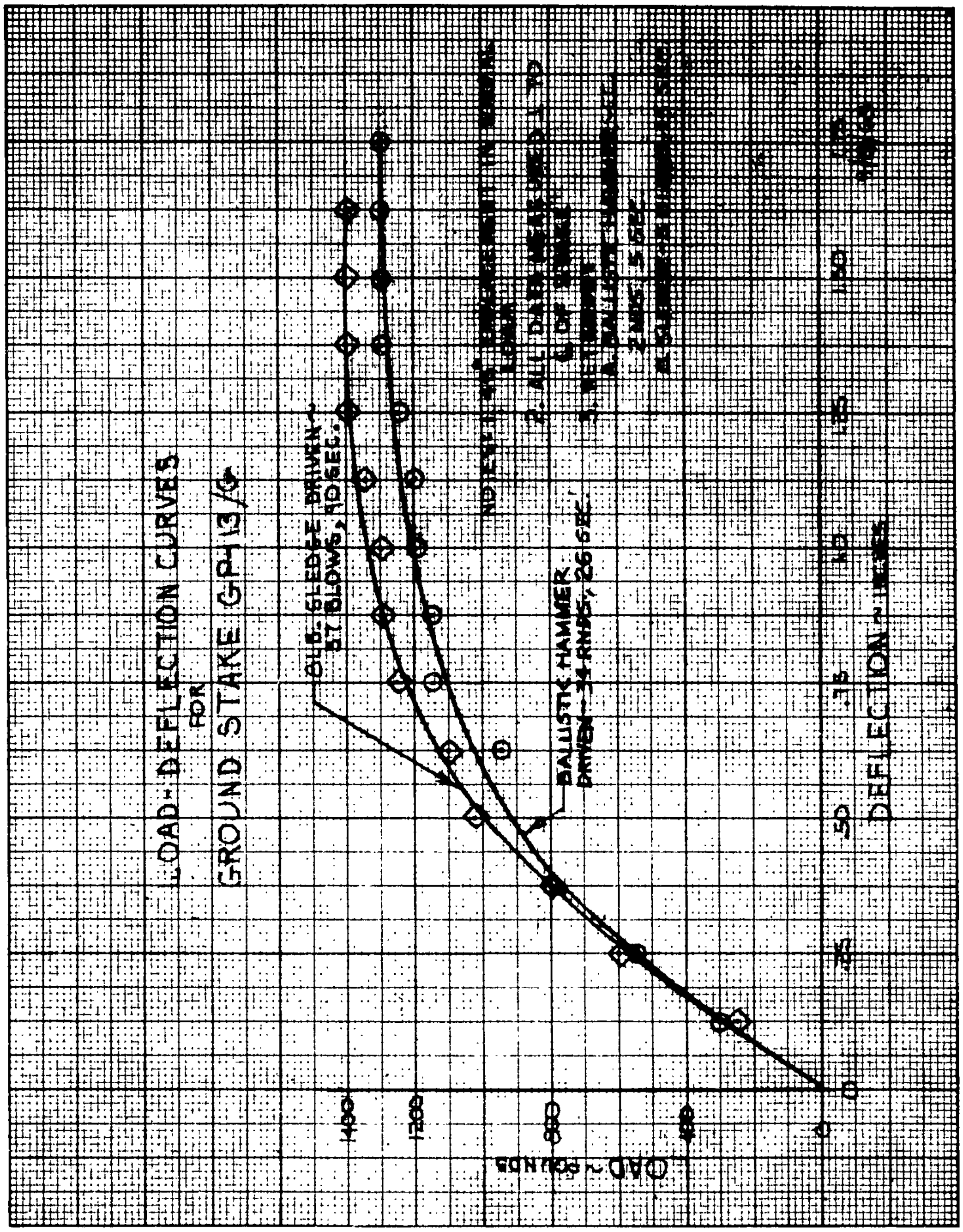

LOAD-DEFLECTION CURVES
FOR
GROUND STAKE GP-13/G
8 LB. SLEDGE DRIVEN-
37 BLOWS, 40 SEC.
BALLISTIC HAMMER
DRIVEN-34 RNDS, 26 SEC.
LOAD ~ POUNDS
DEFLECTION ~ INCHES

Figure 42

Test	Emplacement Technique	Time*	Rounds or Blows	Retrieve Technique	Rounds or Blows	Time*	Maximum Holding Power	Deflection at Max. Load*	Soil Description
1	Tool**	30	13	Hand	--	--	--	--	Loose Sand
2	Sledge	90	33	Tool	1	--	--	--	Loose Sand
3	Tool	60	32	Truck	--	--	700	1-3/4	Compact Sand
4	Sledge	240	66	Truck	--	--	900	1	Compact Sand
5	Tool	--	--	--	--	--	1825	7/8	Clay
6	Sledge	--	--	--	--	--	2100	3/4	Clay
7	Sledge	195	116	Tool	4	--	1825	1-1/4	Loam, Clay
8	Tool	40	48	Tool	4	5	1750	1-1/4	Loam, Clay
9	Tool	9	7	Hand	--	--	525	1-3/8	Moist Sand
10	Sledge	25	16	Sledge	12	30	725	1-1/2	Moist Sand
11	Tool	10	12	Tool	6	5	650	1	Moist Sand
12	Tool	7	9	Hand	--	--	--	--	Moist Sand
13	Sledge	25	16	Hand	--	--	--	--	Moist Sand
14	Tool	24	30	Sledge	6	10	1750	1-1/2	Manor Loam
15	Sledge	70	42	Sledge	6	10	1700	1-5/8	Manor Loam
16	Tool	24	32	Tool	3	5	2000	1-7/8	Manor Loam
17	Sledge	90	56	Hand	--	--	2000	2-1/4	Manor Loam
18	Tool	26	34	Tool	2	5	1300	1-3/4	Manor Loam
19	Sledge	90	57	Tool	3	15	1500	1-5/8	Manor Loam

* Times shown are actual working times and do not include rest periods, time to emplace clips or clearing a jam. All times are recorded in seconds, holding power in pounds and deflection in inches. Tests 3,4,18, and 19 were conducted with stakes driven at 45° with respect to the ground; in all others the stakes were driven at 90° with respect to the ground.

** Tool refers to Ballistic Hammer MX-6321 ( )/G

Summary of Pull Test Results
Figure 43

"1-(c)" and "1-(d)"*, it represents an average of a minimum of ten (10) individual tests. Emplacement was performed at angles from 30° with the ground to perpendicular. No noticable difference in emplacement time was noted.

The data is presented in such a manner that a performance comparison may be held between the Device and sledge. The following facts are pertinent to this comparison:

a. Momentum and Energy Delivery

	Hammer Weight (lb.)	Hammer Velocity (vt/sec.)	MV (lb-sec.)	KE (ft-lb.)
Sledge Hammer	8	32	8	130
Ballistic Hammer	4	116	14.5	845

b. Times indicated are actual working times and do not include rest periods, initial positioning blows (applicable to sledge only), and time to emplace clips or clear a jam.

c. For Ground Stake GP-113/G, a verbal soil description relevant to SCL-4359 is included as are torque-penetration data taken with the A.B. Chance Soil Evaluation tool.

(1) <u>Ground Stake GP-112/G</u>

(a) Soft Stone

Tool ; 23 rounds, 35 seconds

Sledge; 55 impacts, 100 seconds

(b) Solid, Hard Ice

Tool ; 12 rounds, 13 seconds

Sledge; 16 impacts, 30 seconds

(c) Frozen Ground)

(d) Hard Pan)Permafrost conditioned at -80°F
)

Tool ; 17 rounds, 21 seconds

Sledge; 35 impacts, 80 seconds

* The data for the Device under these situations were obtained with the mock-up device which possessed less energy and momentum delivery than the actual device. Respective times indicated are extrapolated.

(2) Ground Stake GP-113/G

(a) Crumbling, damp

(Class 4) principally clay in a state that would crumble when an attempt was made to squeeze it into a ball with the hand.

Torque (in-lb)*	Penetration (ft.)
100	1
225	2
275	3
275	3-1/2

Tool ; 31 rounds, 27 seconds

Sledge; 65 impacts, 110 seconds

(b) Firm, moist

(Class 5) clay and sand predominant, when squeezed with hand, it formed firm ball.

Torque (in-lb)*	Penetration (ft)
125	1
325	2
500**	2-1/2

Tool ; 34 rounds, 28 seconds

Sledge; 60 impacts, 95 seconds

(c) Hard, dry

Principally clay in a state of hardness that approaches that of Hard Pan (Class 3)

Torque (in-lb)*	Penetration (ft)
225	1
325	2
400	3
400	3-1/2

Tool ; 46 rounds, 40 seconds

Sledge; 118 impacts, 210 seconds

---

*Measured using A.B. Chance soil evaluation tool.

**Probable strata of gravel. No higher readings possible with torque wrench.

(d) Loose, dry

(Class 7) predominated by sand and gravel - soil remained very loose due to lack of bonding material to hold the particles together.

((1)) Tool ; 13 rounds, 30 seconds

((2)) Sledge; 33 impacts, 90 seconds

(e) Compacted, moist

Predominated by sand an gravel; consistency resembled Class 6. Soil was compressed by surface loading.

Torque (in-lb)*	Penetration (ft)
75	1
175	2
175	3
175	3-1/2

((1)) Tool ; 32 rounds, 65 seconds

((2)) Sledge; 68 impacts, 250 seconds

(f) Loose, wet

(Class 7) contained wet sand, loam, and silt. Resembled mud-free water drained into open hole.

Torque (in-lb)*	Penetration (ft)
50	1
75	2
100	3
100	3-1/2

((1)) Tool ; 8 rounds, 10 seconds

((2)) Sledge; 16 impacts, 25 seconds

5. Retrieve Capability

The developmental retrieve tests and analytical investigations have been discussed previously. The data presented on the following page has

been established for the Device in the "as-delivered" condition. It has been arranged to coincide with the previous discussion relating to the emplacement capabilities of the Device. This data is for the most critical retrieve situation, i.e., when the stakes have not been loosened in the ground by any external loadings. The remarks previously made regarding time data, particular soil types, and test conditions apply. All data below indicates the impacts, and time to impart impacts, in order that the stake be sufficiently loose to enable retrieval by hand.

a. <u>Ground Stake GP-112/G</u>

(1) Soft Stone

Tool ; 6 rounds, 15 seconds

Sledge; 90 impacts, 220 seconds

(2) Solid hard ice

Tool ; 3 rounds, 7 seconds

Sledge; 15 impacts, 40 seconds

(3) Frozen ground }

(4) Hard pan } Permafrost conditioned to -80°F *

Tool ; 5 rounds, 13 seconds

Sledge; 30 impacts, 75 seconds

b. <u>Ground Stake GP-113/G</u>

(1) Crumbling, damp

Tool ; 4 rounds, 10 seconds

Sledge; 20 impacts, 50 seconds

(2) Firm, moist

Tool ; 4 rounds, 10 seconds

Sledge; 25 impacts, 60 seconds

(3) Hard, dry

Tool ; 7 rounds, 12 seconds

Sledge; 50 impacts, 110 seconds

---

* The data for the device under these situations were obtained with the mock-up device which possessed the same operating principle as the "delivered" device. Respective times indicated are extrapolated.

(4) Loose, dry
   (a) Tool ; 1 round, 3 seconds
   (b) Sledge; 6 impacts, 15 seconds

(5) Compacted, moist
   (a) Tool ; 4 rounds, 9 seconds
   (b) Sledge; 30 impacts, 70 seconds

(6) Loose, wet
   Withdrawn by hand in both instances.

It should be noted that the times are average and do not include several instances where the sledge technique did not permit the stake to be withdrawn.

6. Stake Life

A very important consideration with respect to the Ballistic Hammer is its effect on the "life expectancy" of the stakes concerned. In the previous operation involving the sledge hammer, stakes were retrieved by blows applied perpendicular to the longitudinal axis of the stake. With the Ballistic Hammer, however, stakes are retrieved by blows applied parallel to their longitudinal axis. The stakes were not expressly designed to resist this manner of loading. In addition to this, the Ballistic Hammer imparts higher loadings upon the stakes during the emplacement operation than does the sledge. For these reasons, a program was conducted to study the effects of these considerations on the stakes.

The table below relates the momentum and energy of the 8 pound sledge and Ballistic Hammer:

	Hammer Weight (lb.)	Hammer Velocity (ft./sec.)	MV (lb-sec)	KE (ft-lb)
Sledge Hammer	8	32	8	130
Ballistic Hammer	4	116	14.5	845

a. Ground Stake GP-112/G

With this stake, no damage was observed after as many as 15 emplacement and retrieve operations with both the Ballistic Hammer and the 8 pound sledge hammer. The only signs that the stakes had actually been used were scraping of the painted surface and some peening of the area of the stake which was impacted by the emplacement means. The peening was more severe on the sledge-driven stakes, but was not of such a degree as to effect the operation of the stake to any degree.

b. Ground Stake GP-113/G

(1) Welded Design

In this design the impact head of the stake is joined to the body by welding the two together around the circumference of the tube end and by two holes drilled through the tube such that the tube and the head body can be welded together. If this stake is emplaced a sufficient number of times by any means, the tube will begin to crack in the location of the welded holes due to their induced stress concentrations. As indicated in the table below, there is some increase in the stake life if it is emplaced and retrieved by the sledge.

	Emplacement-Retrieve Cycles Before Failure*	Emplacement-Retrieve Cycles Before Failure (all stakes sledge-retrieved)*
Sledge Hammer	15	15
Ballistic Hammer	9	12

The above data indicates that the effect on the stake life is not strongly affected by the Device. The reason for this is not readily apparent, since the Device delivers approximately twice the impulse and six times the energy of the sledge. One important consideration, however, in evaluating this data is that the stakes are continuously being cycled in their plastic region of deformation with both emplacement techniques.

* See Footnote on the following page.

(2) Swaged Design

The general design features of this are the same as for the welded design except for the holes in the tube to permit plug welding to join it with the head. In place of this is a wide circular groove in that part of the head which lies in the tube. The tube is then swaged into this groove to rigidly join the two. This design appeared to have little effect in eliminating the splitting of the tube during emplacement. During emplacement, the emplacement means tends to push the head down the tube until the groove engages the swaged area very tightly, resulting in a very effective connection. During Device retrieval, however, the head moves in the other direction, resulting in the swaging being nearly completely ineffective - the result is that the weld must carry the complete retrieval impact and failure of this weld results after a sufficient number of cycles. If the weld does not fail upon retrieval, the succeeding emplacement blows tend to impact the groove and swaging which results in a wrinkling failure of the tube in the swaged area. The data in the table below indicates this tendency.

	Emplacement-Retrieve Cycle Before Failure*	Emplacement-Retrieve Cycle Before Failure (all stakes sledge-retrieved)*
Sledge Hammer	10	10
Ballistic Hammer	6	12

From this data it can be seen that the welded design is slightly superior to the swaged design. It should be re-emphasized that in the field operation of the overall Device, the stakes which will be retrieved will have been loosened in the ground by their applied loading.

* This represents complete perpendicular emplacement and the associated retrieve operation in hard, dry clay. Typical data relating to the characteristics of this soil is presented below:

	Emplacement Blows	Retrieve Blows
Sledge Hammer	116	64
Ballistic Hammer	50	7

Under this condition, considerably longer life would be experienced with the Device-retrieved stake.

It should be noted that there were instances where stakes of both designs failed after as few as 100 blows were applied. There was a noticeable lack of uniformity in the performance and appearance of the stakes and it would appear that a stake redesign should be considered if the potential of the ballistic emplacement-retrieve approach is to be fully realized.

7. Strain Gage Investigation

During the course of the program, several instances were encountered in the structural design of the Device for which theoretical analysis could only give a first approximation as to the structural integrity and response of the components considered. For this reason, strain gage investigations were conducted in a number of instances:

a. The stresses in the Barrel under conditions of a "no-load" firing (the Hammer impacts the Barrel's Buffer System under this condition) were investigated. This investigation revealed that the resultant stresses in the Barrel were lower than analytically predicted. This resulted in enabling the removal of a portion of the wall of the Barrel to a thickness of 1/8 inch. Further strain gage investigations revealed that the stress in the modified Barrel was 68,000 psi, which was suitably below the yield stress of the Barrel material.

b. The critically stressed areas of the Retrieve Device were analyzed using strain gage techniques. A detailed discussion of this analysis is found in the stress analysis for this device.

It should be mentioned that a strain gage investigation was conducted on Ground Stake GP-113/G both under impact from the Device and from a sledge. Investigation of these results revealed that in some instances the strain gages indicated that the material was undergoing plastic deformation On this basis all previous strain gage data for the stakes were considered to have questionable value.

Because of the above, a device was designed to measure the relative magnitudes of the two blows. This consisted of a reservoir containing hydraulic fluid with a piston in one end. This device was placed on the head of the stake. The piston was then impacted by the emplacement means (Device or sledge) which generated a pressure in the hydraulic fluid. This pressure was measured by a piezoelectric pickup and associated instrumentation. While the data obtained was of a qualitative nature it indicated that the Device imparts a slightly higher force upon the stake than the sledge.

8. Drop and Jumble Tests

While no formal evaluation of the resistance of the Ballistic Hammer to drop and jumble situations was conducted, these considerations were attended to during the development of the Device.

a. Jumble

In the course of its development, much testing of the Device occurred at sites far removed from the main building areas of AAI. While traveling to these sites, the Device was placed unprotected (not in the Tool Roll) in the metal bed of a pick-up truck and transported in this manner to the site. In most instances, no roads led to the sites - the journey took place "across country" over very rough terrain. The total distance over which the Device was transported in this manner was approximately 75 miles. No damage of any extent was incurred by any component.

Other journeys took place under the same vehicular conditions over primary and secondary roads at legal speeds. The total distance traveled under these conditions was approximately 100 miles. Again, the Device incurred no damage of any degree.

b. Drop

Again, during the developmental program the Device was accidentally dropped onto every type of soil condition under completely random orientations. Height of drops ranged to six feet. No damage was incurred.

9. Effects of Soil Environment

This consideration covers the effects of the soil environment as far as the soil actually interfering with the normal operation of the Device is concerned. In normal operation of the Device and also normal conditions of laying the Device upon the soil, this is no problem. If the Device were actually buried in sand, (for example) however, it is very likely that it would clog the area of the Hammer Return Assembly. If it were possible to fire the Device under this condition, the presence of the sand would probably not damage any components, but wear would be induced at an accelerated rate. At any rate, it would probably be impossible to fire the Device since the carbine action would not be expected to feed any cartridges or cycle properly.

10. High Speed Movie Tests

Many tests employing high speed photographic techniques were conducted during the course of the program. The areas of particular interest are discussed below:

a. The carbine action cycle was studied. This was done in order to study operational malfunctions during the early stages of the program. The mean operating slide velocity of the action was found from this data to be 15 ft/sec.

b. The Hammer velocity was experimentally established as 116 ft/sec. (The analytically determined velocity was 107 ft/sec).

c. The velocity of the recoiling mass of the Device during emplacement was found to be 14 ft/sec*. The analytical recoil velocity is found to be:

$$m_H V_H = M_\rho V_\rho$$

$$V_\rho = \frac{(M_H)}{(M_\rho)} V_H$$

$$= \frac{4}{30.95} \quad 116$$

$$V_D = 15 \text{ ft/sec.}$$

These two velocities coincide very closely.

* This is a mean velocity of over a 5 inch recoil distance while emplacement of Ground Stake GP-113/G was being undertaken.

V. STAKE PENETRATION EFFICIENCY

After a reasonable amount of testing had been completed, it became apparent that it would be desirable to ultimately achieve stake penetration with expenditure of fewer cartridges.

Basically, stakes are driven into a resistive medium through a very small but finite time of momentum exchange between the driver and the stake. Regardless of the mechanism of penetration and all other parameters being equal, increased penetration will occur with increased stake momentum. This means that identical stakes in identical soils and at the same depth of penetration will be driven further per blow if the velocity imparted to the stake is greater.

In an ideal situation where no permanent deformation or heat losses occur, two equations govern the mechanics of impact -- energy and momentum. By virtue of its mass and velocity, a driver has a measurable amount of energy and momentum before impact that must be conserved after impact. Hence it follows that:

Conservation of Energy (no plastic strain or heat losses)

$$1/2\ M_D\ V_D^2 = 1/2\ M_D\ V_{D'}^2 + M_s\ V_s^2 \qquad (1)$$

Conservation of Momentum:

$$M_D V_D = M_D V_{D'} + M_s\ V_s \qquad (2)$$

where:
- M_D: mass of driver; slugs
- M_s: mass of stake; slugs
- V_D: driver velocity before impact; ft/sec
- $V_{D'}$: driver velocity after impact; ft/sec.
- V_s: stake velocity after impact; ft/sec

Certain momentum and energy losses occur which do not appear in equations (1) and (2). This is a result of the frictional forces in the

medium acting during driver-stake impact. However, since the force of impact is usually considered greater than the frictional resistance, the frictional impulse and energy loss are neglected during impact thus validating equations (1) and (2).

Equations (1) and (2) may be rewritten:

$$V_D^2 = V_{D'}^2 + \frac{M_s V_s^2}{M_D} \tag{1.1}$$

$$V_D = V_{D'} + \frac{M_s V_s}{M_D} \tag{2.1}$$

Eliminating $V_{D'}$ and solving for V_s:

$$V_s = \frac{2 M_D V_D}{M_D + M_s} \tag{3}$$

Equation (3) clearly illustrates the velocity imparted to the stake as a function of the mass of the driver and stake and the velocity of the driver. While the stake velocity varies as the driver velocity, it also varies as the driver mass to driver plus stake mass ratio. Increasing either the driver mass or velocity will increase the stake velocity and hence increase the penetration under similar stake conditions. The tests conducted here were an attempt to verify the validity of this analysis.

During the early phases of the program some experimentation had been conducted to determine the energy required to drive the Stakes GP-113/G and GP-112/G, however, it was not until sufficient test data, collected as the result of runs with the engineering model, was available that the significance of the momentum relationship was more apparent. It should be noted that all work in the early phases of the program was predicated upon maintaining a total weight of 25 pounds, the hammer weight and velocity employed in the engineering model appeared to produce the maximum attainable results within those limits. The possibility of conducting tests to

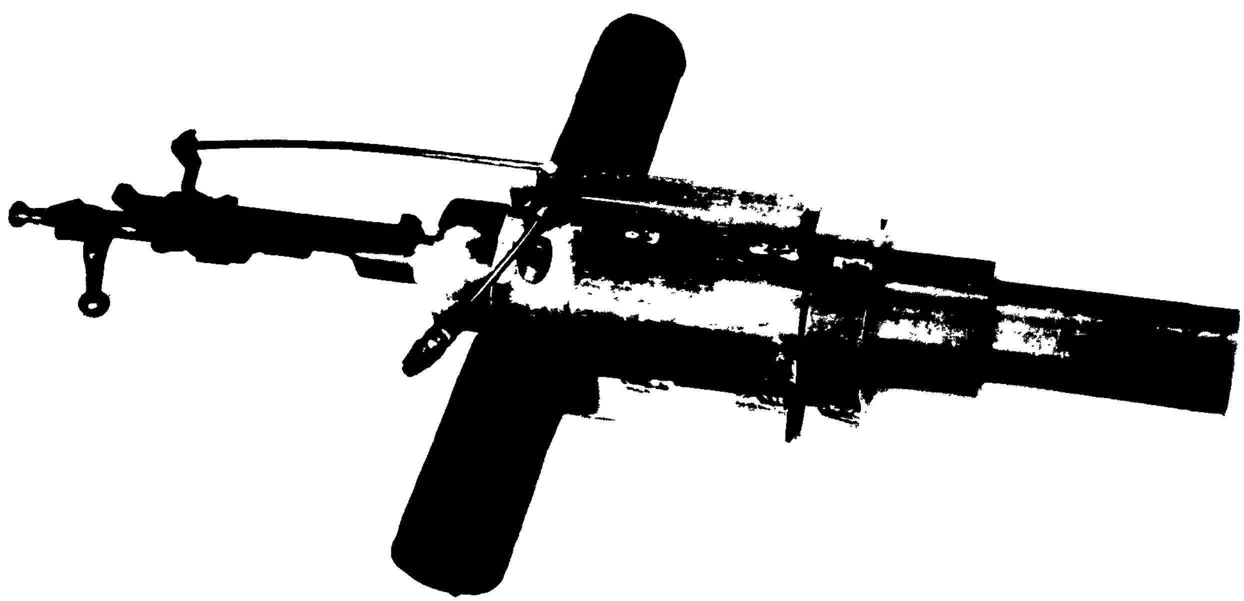
investigate the relation of hammer momentum to stake penetration was discussed with USAELRDL and as a result instructions were formulated requesting AAI to conduct tests.

Tests were all conducted with the breadboard driver employed in some of the early phases of the program. Two pistons were fabricated, one weighing four pounds, and one weighing eight pounds. A jacket was fabricated to permit the total breadboard weight to be increased to fifty pounds. Figure 44 illustrates the breadboard with jacket as employed in these tests.

Since the degree of operator recoil which would be tolerable must be a controlling factor for the device, it was decided to assemble the breadboard to its 50 pound weight with the 8-pound piston and slowly increase hammer velocity by increasing the propellant charge in the cartridge one grain at a time. Results of this test are shown below:

Stake GP-113/G, Soil - Clay

Shot	Charge Weight (grains)	Recoil (inches)	Remarks
1	6	1	
2	7	1	
3	8	2	
4	9	3	
5	10	4	
6	11	5	
7	12	6	
8	13	7	
9	14	9	
10	15	10-1/2	
11	16	12	
12	17	14	Same as Engineering model
13	17	14	"
14	17	14	"
15	17	14	"



Breadboard Model of Ballistic Hammer with Jacket

Figure 44

This indicated that a tolerable recoil would be experienced with the tool weighted to 50 pounds and the hammer momentum twice the value employed in the engineering model.

To verify this performance in stone, a test series was conducted with the tool weighted to 50 pounds, using a 17-grain propellant charge. Recoil of 14 inches maximum was obtained in a group of six shots.

The next test conducted reduced the tool weight to 25 pounds, all other conditions being as described previously. Since an excessive recoil was predicted analytically in this case, the tool was suspended from a fixture rather than being hand held. When actuated the tool bounded approximately five feet in the air which would obviously be excessive. A second test was conducted with the tool weighted to 45 pounds (5 pound reduction), the tool bounded approximately 16 inches, however the operator "felt" considerably more load than had been noted when the tool was weighted to 50 pounds. It may be concluded from this group of tests that, within the range of weights tested, the tolerable operator response is primarily a function of velocity rather than momentum.

Having established that the momentum imparted to the stake could be achieved with a hand held tool, the remainder of the tests were conducted to determine the effect of this increased momentum on stake driving efficiency. It had been observed during the tests to determine recoil that the penetration was increasing proportionally to increased momentum. In increasing propellant charge from 6 to 17 grains penetration per blow increased from 7/8 inch to 3-15/16 inches. (GP-113/G in clay).

A control test was conducted in clay with Stake GP-113/G and the breadboard tool equipped with a 4-pound piston, impact velocity 116 ft/sec. Results are as follows:

Shot No.	Penetration per Blow (inches)
1	1-3/8
2	3/4
3	1-1/4
4	2
5	1-3/4
6	1-3/4
7	1/2
8	1
9	1-1/2
10	1-1/4
11	3/4
12	1
13	3/4
14	3/4
15	1/2
16	1
17	1
18	1/2
19	1
20	5/8
21	5/8
22	7/8
23	3/4
24	1
25	3/4
26	3/8
27	1-1/8
28	3/4
29	5/8
30	3/4

Shot No.	Penetration per Blow (inches)
31	3/4
32	7/8
33	7/8
34	5/8
35	5/8
36	1/2 (Total Penetration - 33)

Under the same conditions the breadboard with an 8-pound piston, impact velocity 116 ft/sec, produced the following results:

Shot No.	Penetration per Blow (inches)
1	4-1/2
2	4-1/8
3	3-1/4
4	3
5	2-1/2
6	2-1/4
7	2-1/4
8	2-1/4
9	2
10	1-3/4
11	2
12	1-3/4
13	1-3/4
14	1-3/4 (Total Penetration - 35.1)

The improvement factor achieved by doubling piston momentum is 34/14 or 2.4.

At this point it was decided to determine that the improved performance was not merely a function of increased piston weight, independent of velocity. To check this the breadboard was assembled with eight pound piston and a 7-grain charge employed to produce an impact velocity of 55 ft/sec.

Penetration of slightly over 1 inch per blow was achieved. This verified the momentum relation as opposed to a purely weight consideration.

Runs were conducted in clay soil during the month of November 1963. Results of a typical run are shown in Figure 45 . Tabulated results are summarized as follows:

	No. of Shots	
Test No.	MV(14.5 lb-sec)	2MV(29 lb-sec)
1	36	12
2	34	14
3	40	15
4	34	14
5	34	14

In the same soil the propellant charge was reduced to 14 grains, this reduced effectiveness as was anticipated.

Shot No.	Penetration per Blow (inches)
1	3-1/2
2	3
3	3
4	2-1/2
5	2-1/2
6	2-1/2
7	2
8	2
9	2
10	1-3/4
11	2-1/4
12	2
13	1-1/2
14	1-1/2
15	1-1/2
16	1-1/2
17	1-1/2 (Total Penetration 36.5)

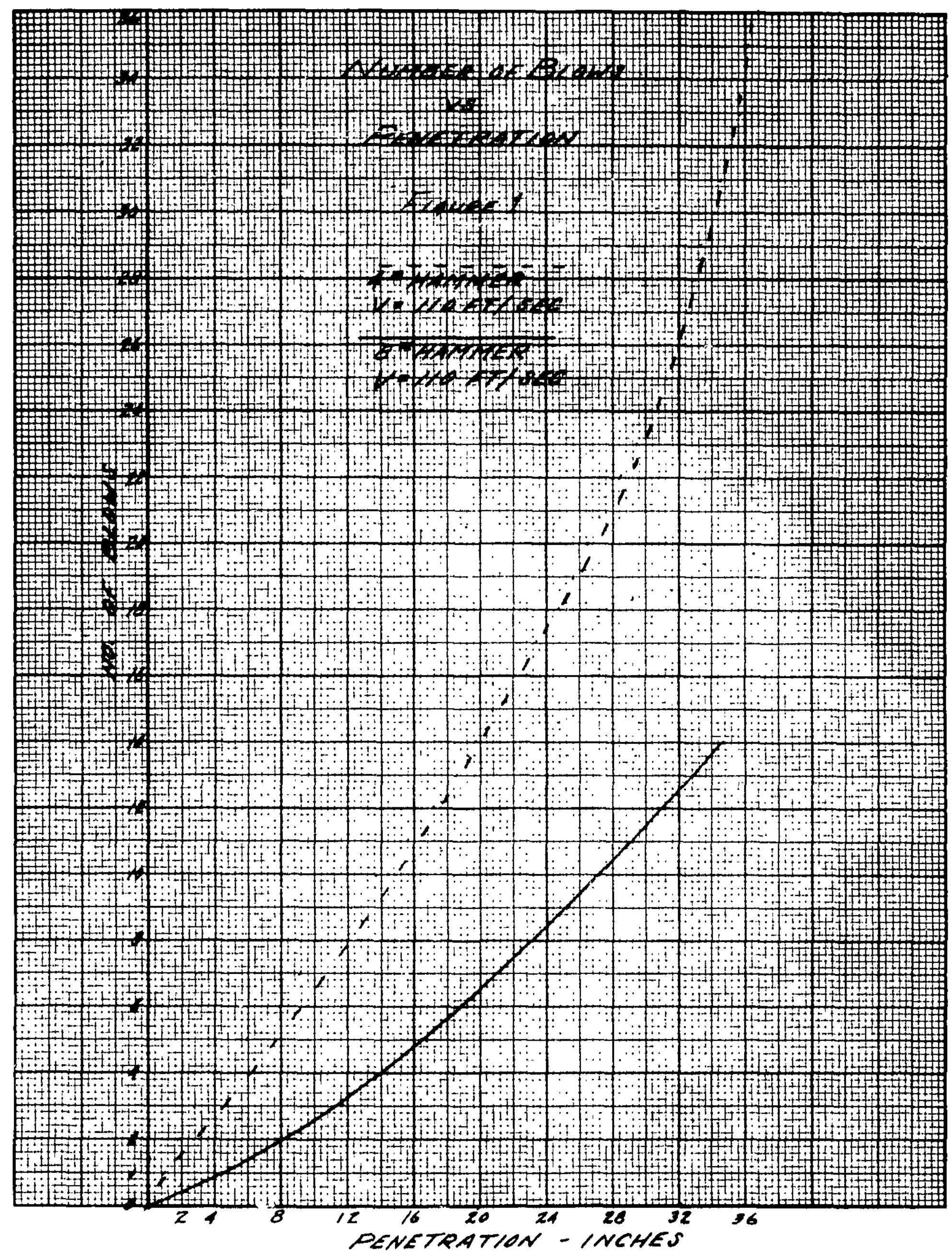

NUMBER OF BLOWS
VS
PENETRATION
FIGURE 1
V = 110 FT/SEC
8" HAMMER
V = 110 FT/SEC
2 4 8 12 16 20 24 28 32 36
PENETRATION - INCHES

Figure 45

A total of 17 blows required compared to 13.8 over the five runs in the same soil obtained when a 17 grain charge was employed.

Runs were also made in other typical soil types, results are tabulated as follows:

Stake	Soil	No. of Blows to Emplace MV(14.5 lb-sec)	No. of Blows to Emplace 2MV(29 lb-sec)
GP-113/G	Loose, wet	8	2
GP-113/G	Firm, moist	31	10
GP-113/G	Firm, sand	34	9
GP-112/G	Soft stone	23	11

In summary, it may be concluded that improvement in stake driving effectiveness in the order of a factor of 3 may be achieved by doubling hammer momentum. In addition, it has been shown that this improvement can be achieved within the framework of a portable man-held tool with a modest weight increase.

VI. DEVELOPMENT HISTORY

The following paragraphs outline the most important phases in the development program which led to the evolution of the design of the Ballistic Hammer. The discussion will be necessarily brief, as only a presentation of the program's "milestones" is intended. Detail description of design features is to be found in other sections of this report.

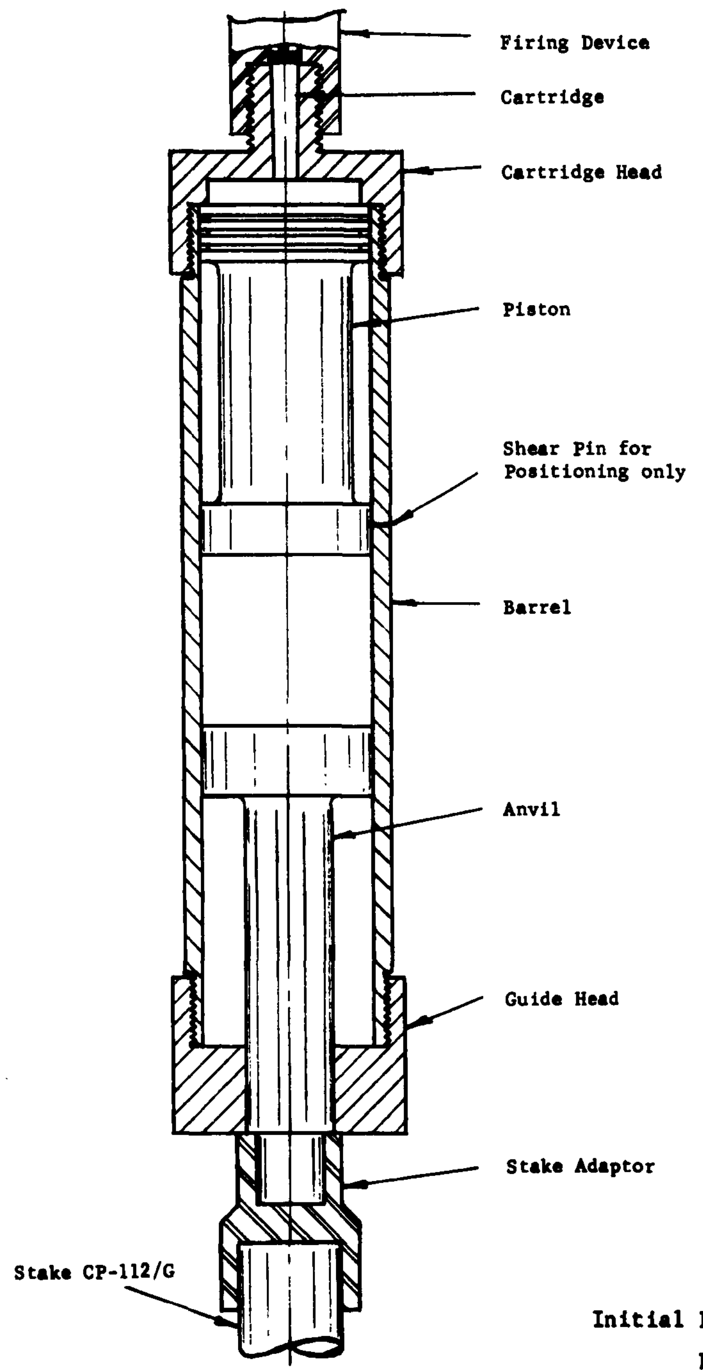
A. Studies With Breadboard Emplacement Device

These were performed to establish the optimum correlation between tolerable recoil impulse, impact energy, impact momentum, and overall tool weight with stake emplacement effectiveness. The governing factors for this with respect to the establishment of the tool design parameters were the 25 pound weight limit imposed upon the device and the tolerable maximum recoil impulse and tool recoil velocity which could comfortably be sustained by the operator.

A layout illustration of the initial breadboard model employed is found on the following pages. This hammer possessed the following features; five (5) pound piston, twenty-five (25) pound overall weight, anvil, barrel, two barrel heads, a single-shot action (.30 caliber) and a stake adaptor. A shear pin was furnished to establish the position of the piston to establish the initial volume for the propellant gases. Tests performed with this device indicated the feasibility of the ballistic device, but improved performance was desired. To achieve this end, handles were installed on the device to enable it to be held and controlled by the operator to the recoil associated with its operation. The lower end of the device was modified to eliminate the anvil of the initial device. (See Figure 30). These resulted in greatly improved performance.

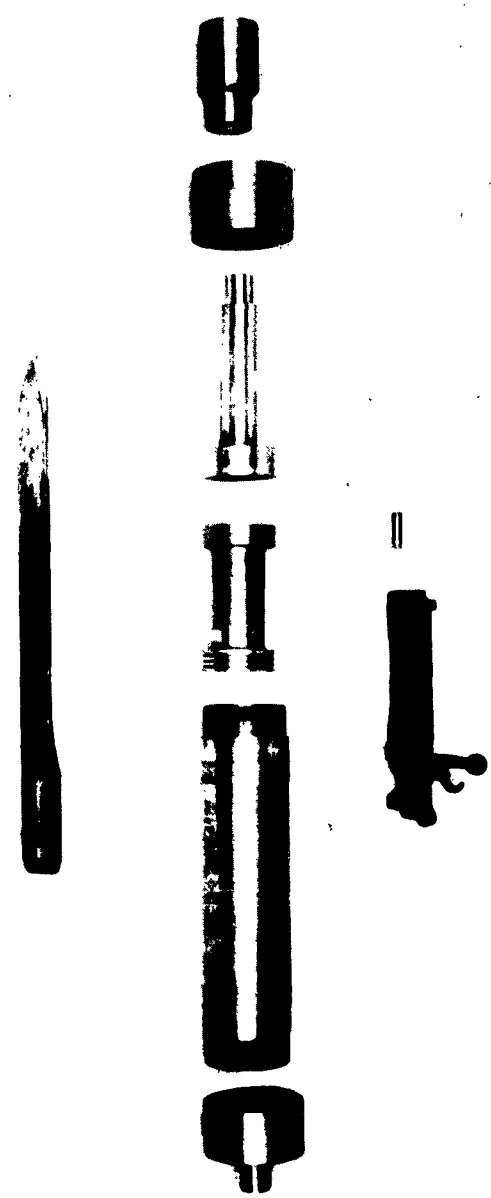
After extensive testing a comfortable recoil impulse of 12.4 lb-sec. (equivalent to a recoil velocity of 16 fps) was established, compared to 8 lb-sec., which was initially determined as the maximum.

B. The establishment of the maximum tolerable recoil level, irrespective of the device used, permitted further design of the final tool configuration.



Initial Ballistic Hammer

Figure 46



Initial Ballistic Hammer Exploded View

Figure 47

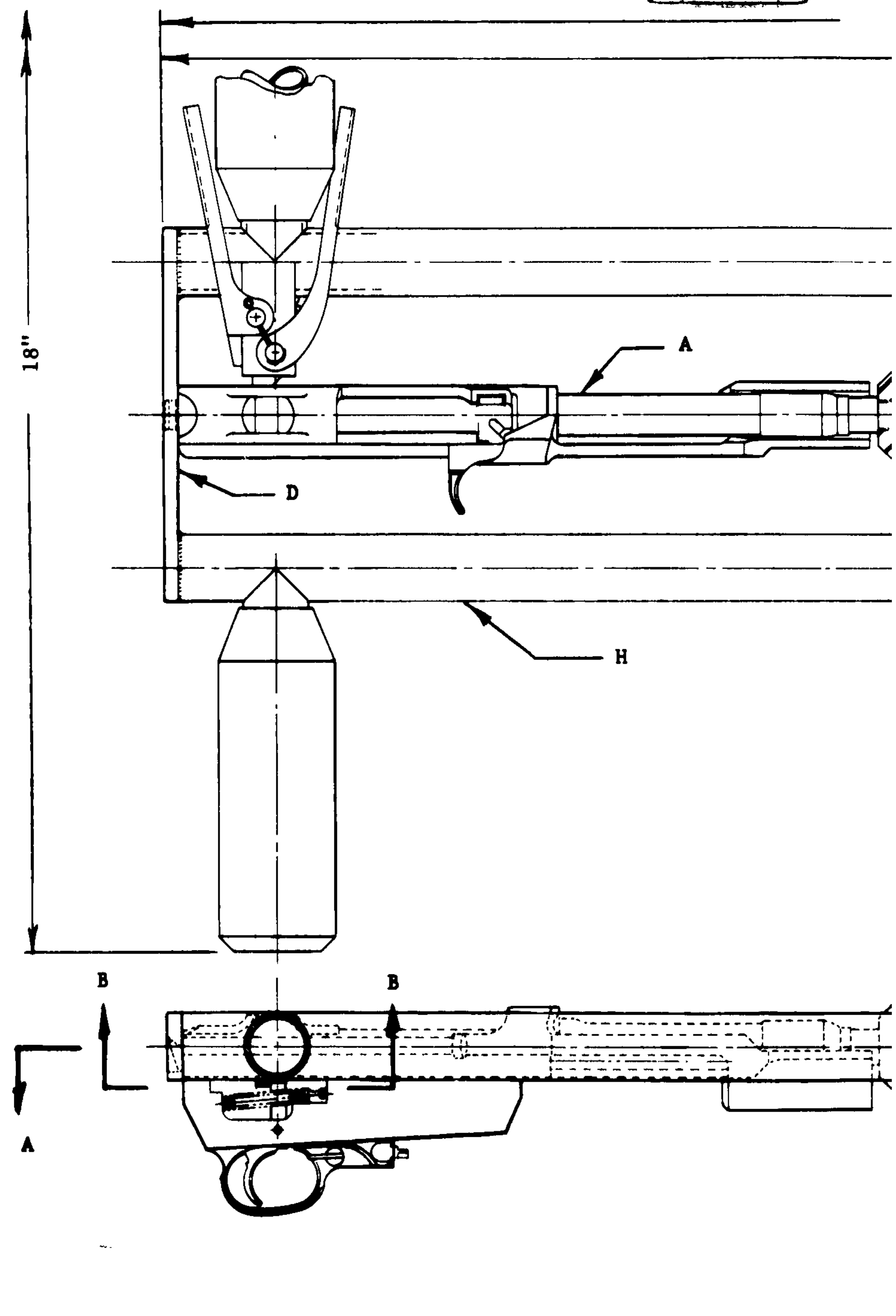
This information resulted in a more sophisticated design approach (refer to the following page) which was presented in the Design Plan (AAI Report No. ER-2937). The emplacement device presented in the Design Plan had the following characteristics: overall tool weight of twenty-nine (29) pounds, semi-automatic caliber .30 carbine action employing grenade launcher cartridges used as energy source, spring returned piston, integrated exhaust-handle system, a positive trigger safety system, and others which would be best found by reference to the layout drawing.

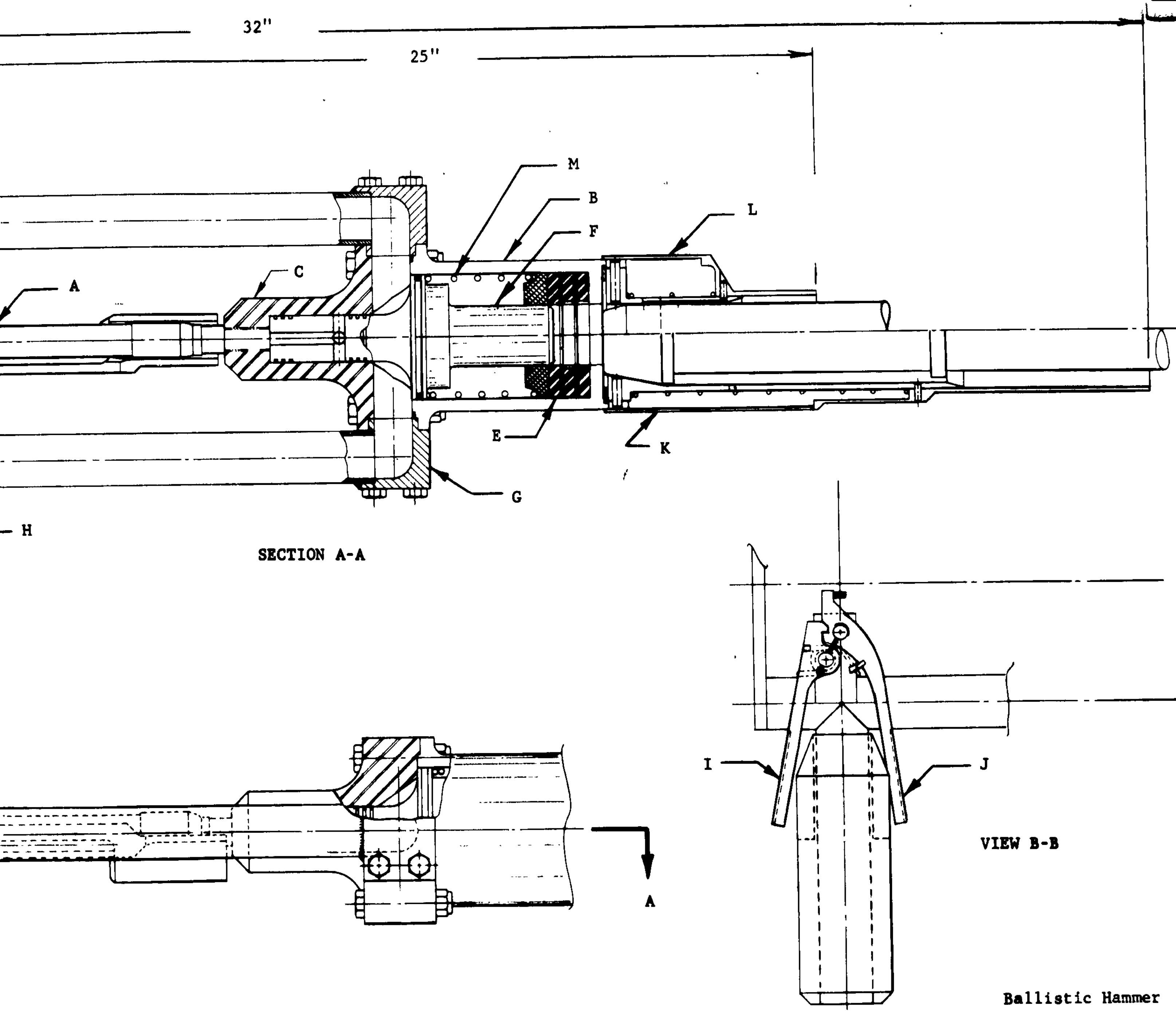
Also presented in the Design Plan was a retrieve approach which offered potential for the rapid retrieval of emplaced stakes and employed the emplacement device as an energy source. The concept selected consisted of connecting the frame of the emplacement device to the stake and preventing the piston from moving relative to the ground by placing a bar between the two. Upon firing, the energy of the propellant gases was used to recoil the frame of the emplacement tool, retrieving the stake. No energy was used to propel the piston since it was "bottomed" against the ground surface.

During the course of experimentation conducted with the tool breadboard, components were employed which duplicated the functions of this retrieve device (see following page). Tests conducted with this device revealed that for either stake no tests conducted required more than three shots (or 2100 ft-lb of energy) to loosen a stake so that it could be withdrawn by hand.

This retrieve approach, as designed to integrate with the Emplacement Device, is shown in Figure 49. A quick connect coupling for attaching the device to the barrel of the Emplacement Device was provided. A cam-type clamp was provided to quickly and easily attach the device to the stake. The base plate assembly shown prevented the piston from moving relative to the ground. Since, upon firing, the coupling clamp assembly moved upward relative to the base plate assembly, a screw adjustment was provided to maintain the initial "ground out" condition of the piston as the stake was retrieved.

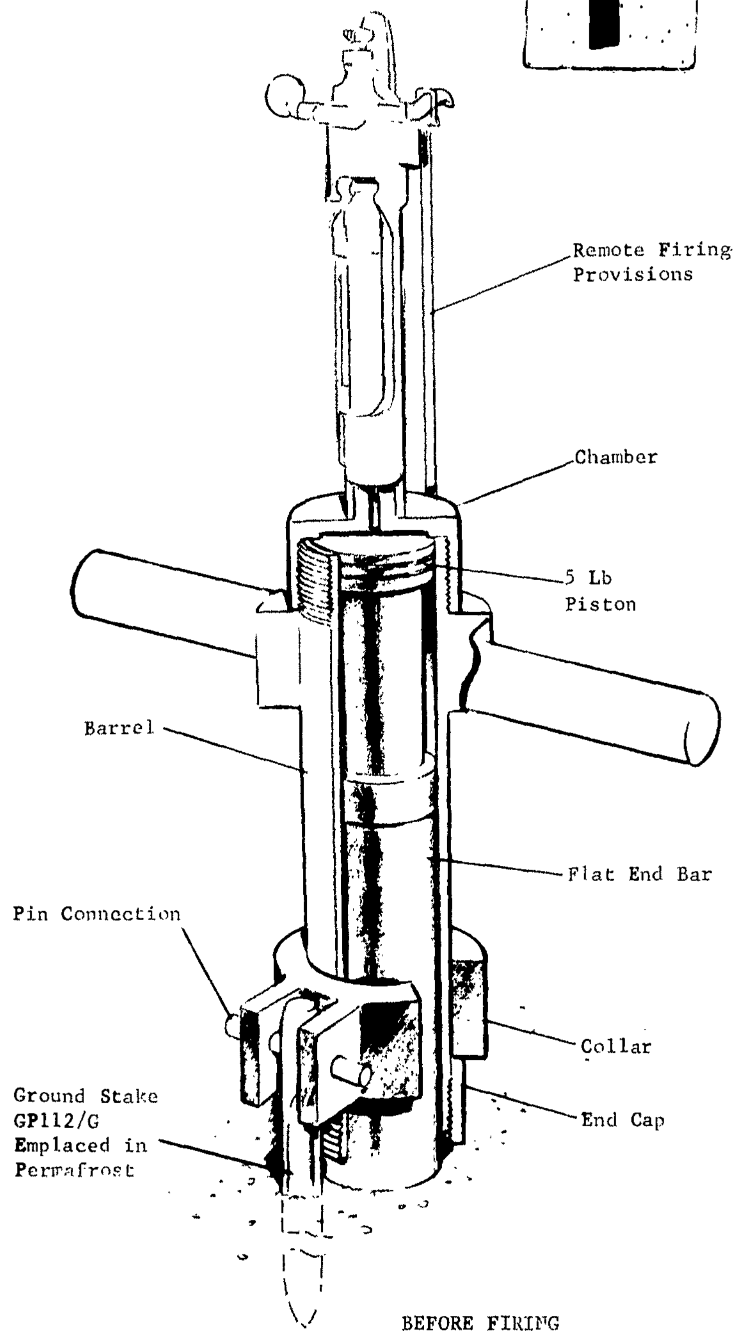
1



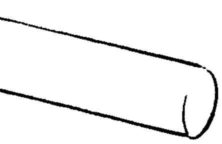


Ballistic Hammer
Figure 48

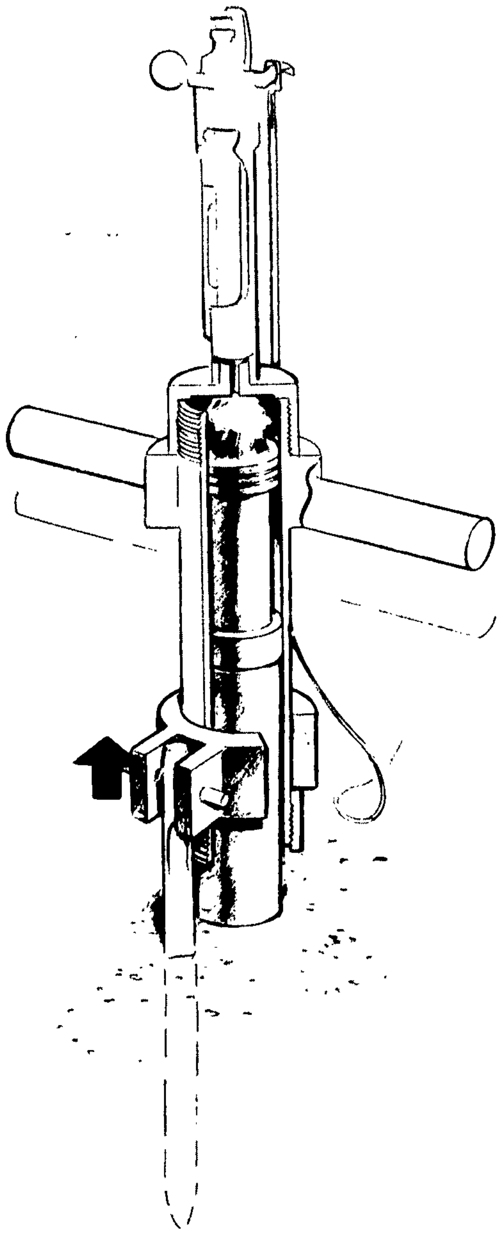
1



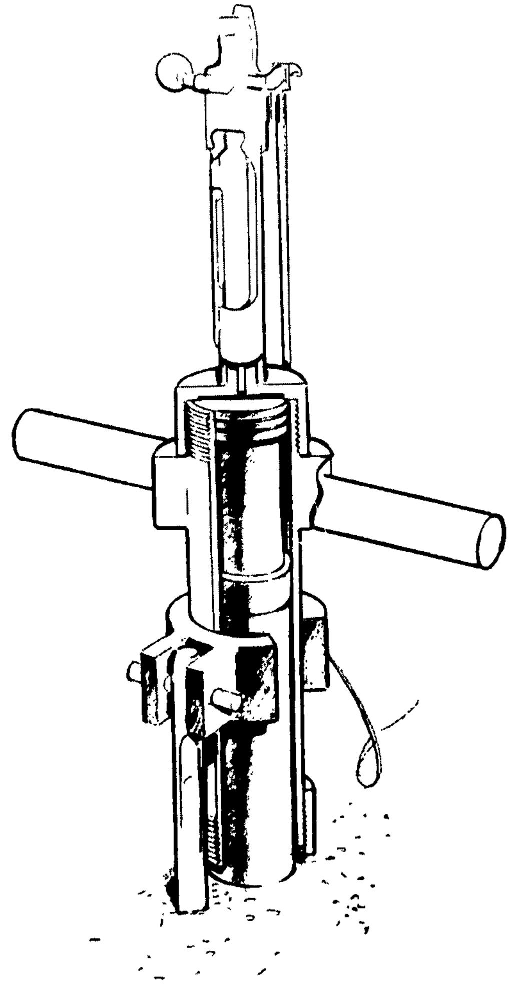
BEFORE FIRING



IRING



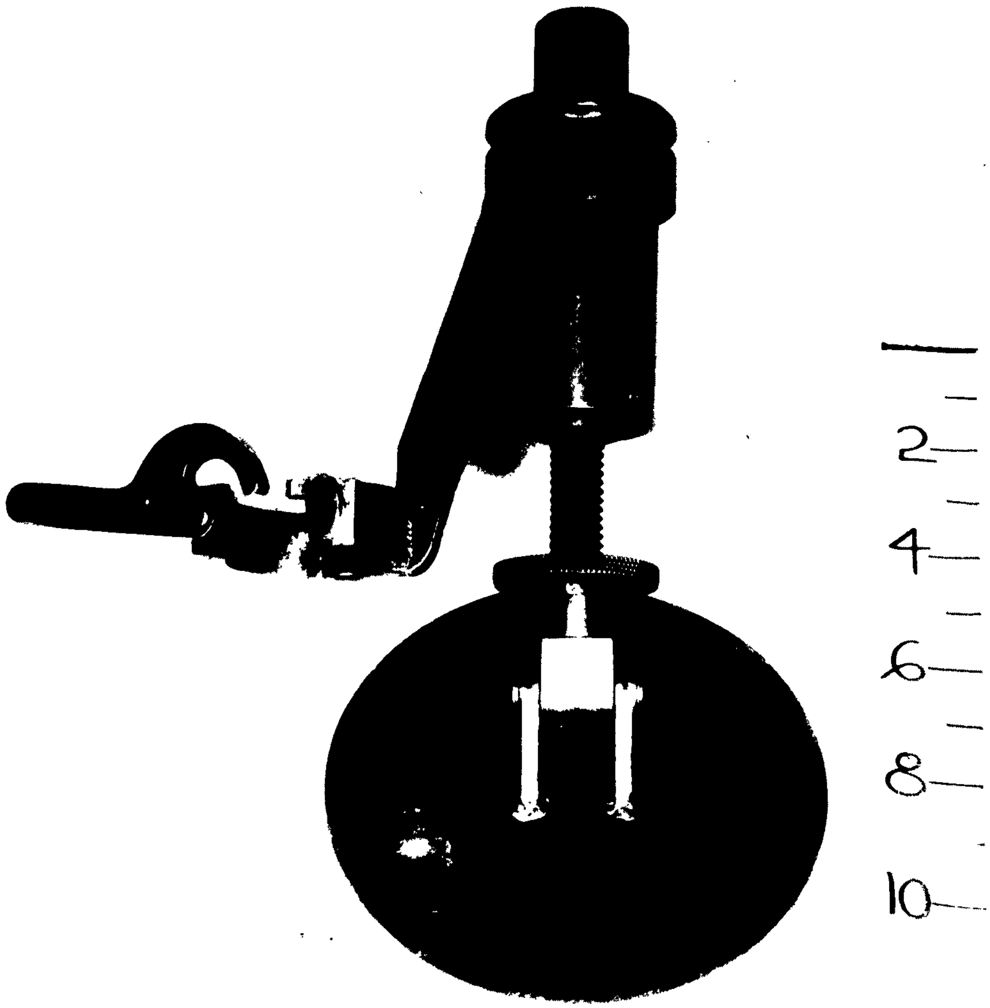
RECOIL EXTRACTION



AFTER RECOIL

Extraction Sequence
(Mockup Retrieve Device Illustrated)

Figure 49



Initial Retrieve Adapter Design

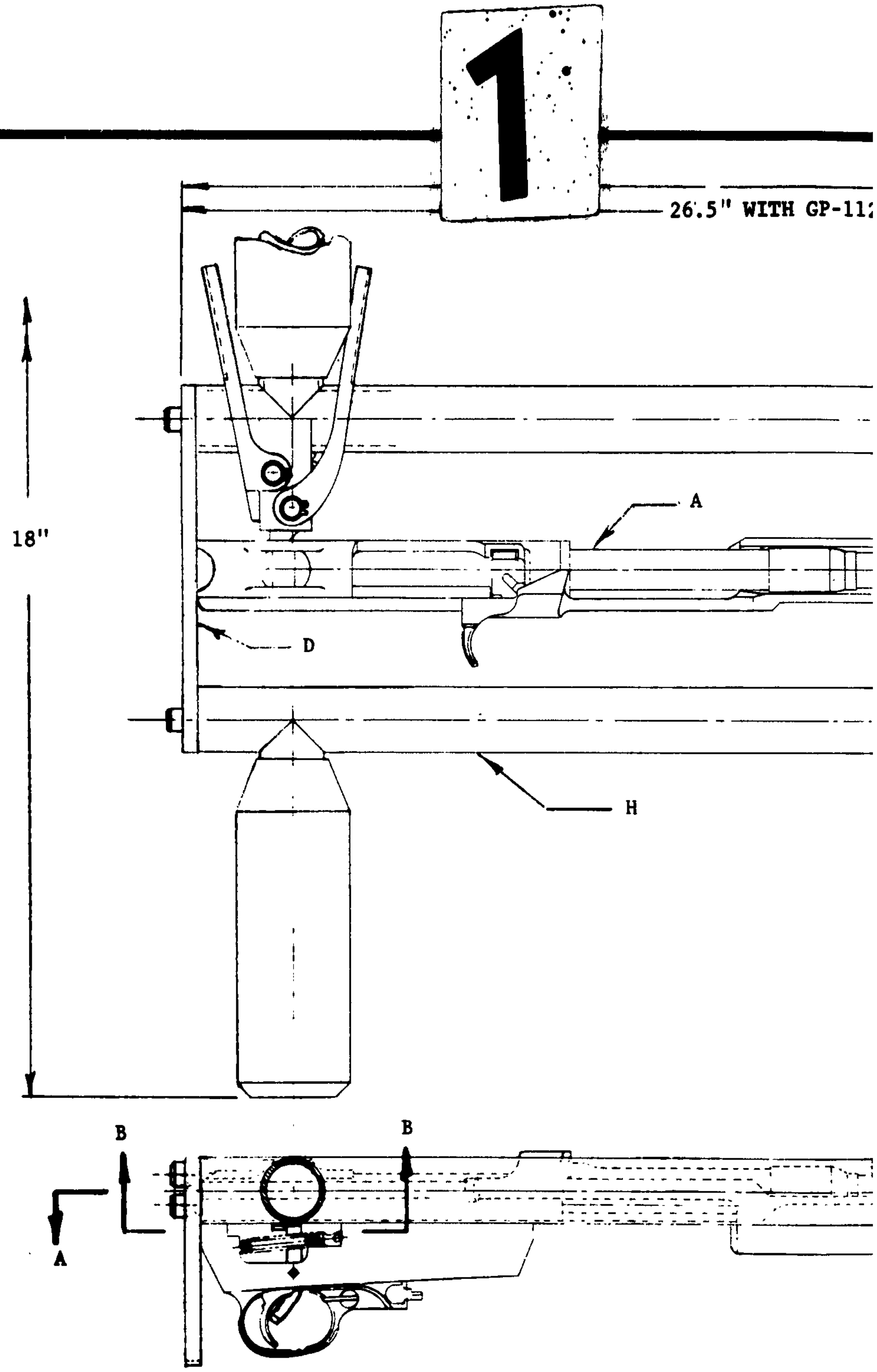
Figure 50

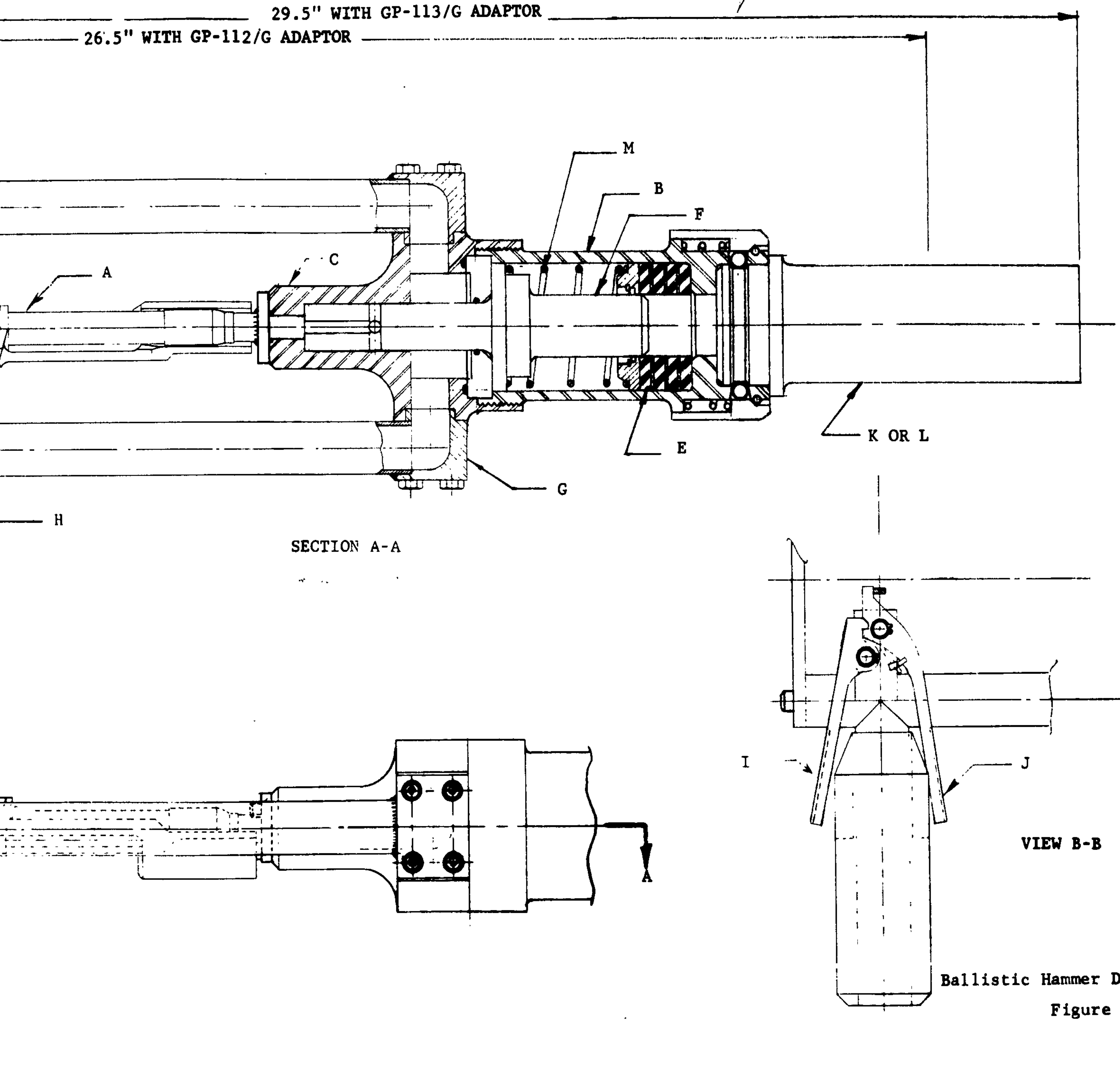
C. Minor revisions of the Design Plan device were instigated before fabrication was begun. The appearance of the revised device can be seen on the following page.

Upon completion of the fabrication of the revised device an extensive test program was started. Test pistons were fabricated weighing three (3) and five (5) pounds. Each piston resulted in 12.4 lb-sec. recoil impulse but delivered different energies. The purpose of firing different piston weights at constant momentum was to determine if increased stake penetration could be obtained with the lighter piston. Analysis of the resulting test data disclosed that the same number of rounds was required to emplace Ground Stake GP 113/G in a very cohesive clay soil, but one-half the rounds were required to emplace Ground Stake GP-112/G in limestone with the three (3) pound piston as compared to the five (5) pound piston. For these reasons the three (3) pound piston was selected. During the above testing phases it was found that two problem areas existed; (1) the bolt velocity of the caliber .30 action was excessive, resulting in bolt-firing pin seizure after approximately 50 shots; and (2) the piston return spring was rendered ineffective after approximately 20 shots. Upon determining this, work was immediately instigated to solve these problems and is described below.

The retrieve adaptor to be attached to the emplacement device was fabricated and evaluation of it started. It was seen that two serious problem areas existed with this design; (1) the device was clumsy to manipulate with regard to fastening its clamp on Ground Stake GP-113/G; and (2) the bolt of the action was cycling under extreme high pressure, resulting in bolt-firing pin seizure. The second problem could not be readily solved without addition of considerable weight. For this reason, major redesign of the retrieve adaptor was begun.

D. Because of inherent simplicity, it was decided that the retrieve adaptor redesign be initially centered around the application of the mockup emplacement device. Several systems were investigated in this manner which again involved connecting the stake to the frame of the tool without "grounding out" the piston. None of these system proved effective or promising. For this


1
26.5" WITH GP-11
18"
A
D
H
B
B
A



Ballistic Hammer Design Layout

Figure 51

reason, thought was given to an entirely new retrieve approach. This finally resulted in the device shown in operation on the following page. This device employed the key lock system to fasten to the emplacement tool. Instead of a stake, the piston impacted the ends of two levers pivoted at their centers, forcing them downward. This caused the other end of the levers to move upward, withdrawing the stake. Initial tests with this device indicated superior performance. Based on this, work was instigated to decrease the device's overall weight and to make it easier to install upon the two stakes. Both objectives were met.

E. At this point work was initiated to improve the overall operation of the emplacement device in the following areas:

1. The feeding problem was thoroughly investigated. It was found that the 30 round magazine was severely damaged by the whiping action of the tool's recoil because of this magazine's length. This led to the use of 15 round magazines. Problems existed with the use of these because the spring rate was not high enough to ensure adequate response during feeding. Substitution of a .046 inch wire diameter magazine spring for the original .031 inch wire diameter spring solved this problem. (It was later found that quantities of magazines were available which were produced with this spring - these were immediately procurred and used). With this spring the performance of the action was greatly improved. A shroud was designed to protect the magazine from damage to its exterior resulting from the whiping action would gradually deform the exterior of the magazine, resulting in the eventual jamming of the follower in the case after approximately 60 shots.

2. While the work discussed in "1" above was being performed, an extensive analysis of the bolt recoil velocity with high speed movies was conducted. From this it was found that the recoil effects were increasing the bolt velocity from what it was with

no recoil. (The frame of the tool being rigidly fixed to a firing table). This excessive velocity resulted in bolt-firing pin seizing after approximately 150 shots. This problem was solved by adjusting the gas piston of the action to decrease the bolt velocity.

3. In parallel to both of the above, high speed movies were taken to determine the causes of random "jams". These disclosed that minor chamber and receiver modifications were needed in the action's barrel so that the grenade launcher cartridges fed more smoothly.

4. Frequent piston breakage (approximately 100 shots) dictated an investigation of this problem area. An extensive analytical analysis disclosed that the cause of this was not strictly one of impact stress but of vibration stress. Upon its impact with the stake, a longitudinal shock wave traveled up and down the piston throughout its length for a great number of cycles. When the wave approached the area of greatest discontinuity the high induced stress concentration was great enough such that, in conjunction with the high frequency of the wave, a fatigue failure occurred. The solution to this problem was to prevent the shock wave from entering the area of greatest discontinuity. This was done by fabricating the piston in two pieces with a rubber shock absorber between the two. (The division between the two pieces occurred at the section of greatest discontinuity).

5. In order to achieve a weight reduction for the overall tool, the possibility of fabricating the entire exhaust system and both (emplacement) stake adaptors from aluminum instead of stainless steel was investigated in detail. This was found to be non-feasible from a stress standpoint due to erosion of the aluminum which occurred due to the action of high temperature propellant gas. Tests conducted in this area demonstrated that this was a satisfactory approach only for the Ground Stake GP-113/G Adaptor.

F. A problem of very significant proportions was that of the piston return spring. Testing of the device revealed that the original design was overstressed due to dynamic effects of high velocity impact and after about 20 shots, the spring could not exert enough force to fully return the piston to its initial position. Because of this, the pressure in the head would not develop to the degree intended. This would result in low piston velocities and, consequently, poor emplacement performance. In addition to this, the action would not cycle properly.

A number of solutions involving many types, configurations, and materials of internal springs were designed and tested. The most successful of these would sustain about 60 shots before the degree of permanent set would seriously affect the performance.

After further analytical analysis, it was found that for this spring the stress was independent of all variable except piston velocity. Thus, for all internal springs the maximum stress induced (psi) was 130 multiplied by the piston velocity in inches per second. In order to maintain the piston design velocity and reduce the spring velocity, it was necessary to isolate the piston and the spring. This was done by connecting the two with a lever system designed to halve the spring velocity. This resulted in a spring system with a theoretical infinite life. Subsequent testing with this design was completely successful.

G. The work relating the Hammer Return Spring and the Hammer breakage resulted in an increase of Hammer weight to four pounds. With the finalized design of the Device, it was calculated that this would result in a recoil impulse of 14.5 lb-sec; higher by over two (2) lb-sec. than that previously found tolerable. Tests subsequently performed demonstrated that the effects on the operator were the same as those when the 12.4 lb-sec. impulse was the criteria. The reason for this was not readily apparant, but as a result of the higher impulse, a further increase in stake penetration per blow was gained.

H. At this stage of the program the design of the emplacement and retrieve device was finalized and the device was performing very consistently -- then the programs enumerated below were conducted:

1. Emplacement-Retrieve Capability

These tests were conducted to concretely establish the capability of the device to perform the assignments for which it was designed. The performance of the 8-pound sledge was compared to the device under each condition. Time and number of impacts were recorded for each condition. In the soils intended for Ground Stake GP-113/G, data was taken with the A. B. Chance Soil Evaluation Tool to specifically identify it.

From these tests it can be said that, in general, for every soil type and each stake, the device required one-half the impacts and one-fourth the time for complete emplacement as the sledge.

2. Holding Power Tests

The immediate objective of these tests was to evaluate any difference in holding power between stakes emplaced (under identical soil and loading conditions) by the Ballistic Hammer and those emplaced by an 8-pound sledge. The concept of these tests was to emplace the stakes in near proximity to each other under identical soil conditions by the two methods. A load, measured by a dynamometer, was then applied perpendicular to the longitudinal axis of the stake by a chain fall. The corresponding absolute deflection of the stake was measured by a pointer attached to it.

The results indicated that, for any particular soil condition, there was little effect on the holding power of the stakes at any stage of deflection induced by the emplacement method.

3. Human Factors

These factors received a high degree of consideration throughout the entire program. The following discussion relates to those of greatest importance considered.

a. Reaction to recoil

As was previously mentioned, the determination of that degree of recoil tolerable to the operator without fatigue was undertaken.

This was found to be 14.5 lb-sec., which corresponds to a recoil velocity of 15 ft/sec. With these parameters absolutely no adverse effects of any type were observed by the operator either immediately after exposure, nor for prolonged periods after exposure.

b. Exhaust gases

At the plane of exit of the Mufflers, these gases possess considerable pressure and velocity, and could possibly contain small particles of unburned propellant or ash, traveling at high velocities. For this reason it was concluded that personnel should come no closer than a distance of six feet from the exit plane of the Mufflers during operation of the Device.

c. Flash and Smoke

(1) Flash

Tests conducted under conditions of reduced visibility demonstrated that there is no flash associated with the Device in any phase of operation, and hence no danger with respect to detection or effects on the operator exist from this source.

(2) Smoke

These conducted in daylight revealed that the smoke associated with the operation of the Device is invisible at approximately 200 feet. It is non-irritating.

d. Control Considerations

The sizes and shapes of the controls of the Device for every phase of operation have been arranged so that the operator has no difficulty in performing these duties with normal or heavy winter clothes on.

e. Noise

This consideration was evaluated by observers stationed at various distances from the tool during operation. At a distance of 700 feet, the tool is virtually silent.

I. Stake Driving Efficiency

As the program progressed, it became possible to determine the number of rounds required to achieve emplacement under the various soil conditions encountered. The number of tests recorded was sufficient to permit a reasonable statistical correlation of the data. Based upon this data it could be concluded that the number of rounds was generally greater than desired from a logistic support standpoint although the number of rounds required could be fired within the desired time for emplacement.

Examination of test results with the tool and comparison with the results of emplacement by 8-pound sledge obtained during time tests, showed a strong dependence of stake penetration per blow upon momentum per blow. This contrasted to earlier considerations where energy per blow appeared a major factor.

This situation was discussed with USAELRDL personnel and it was agreed that it would be desirable to conduct experimental work to determine more fully what relation existed between hammer momentum and stake penetration with a ballistic hammer and to determine whether the momentum levels would be within a range which would be tolerable to a human operator. Instructions formalizing this agreement were received from USAELRDL and the final period of work under the contract was devoted primarily to this effort.

Details of the test conditions and test data obtained are discussed in Section V of this report. General conclusions, however, are summarized below.

Stake driving efficiency was improved by a factor in excess of 3 by doubling hammer momentum in the ballistic model breadboard. This improvement was noted in the complete range of soil types and soft rock evaluated during prior portions of the program. Momentum was doubled by increasing piston weight from 4 to 8 pounds and maintaining the velocity of 116 ft/sec. which had been employed in the engineering model. Summary

of results is tabulated in the following table, blows recorded are the number required for complete emplacement of the particular stake:

Stake	Soil	No. of blows to Emplace MV(14.5 lb-sec)	No. of blows to Emplace 2MV(29 lb-sec)
GP-113/G	Clay approaching hard pan	46	14
GP-113/G	Loose, wet	8	2
GP-113/G	Firm moist	31	10
GP-113/G	Firm sand clay (Fort Monmouth, N.J)	34	9
GP-112/G	Soft stone	23	11

Improvement factor, all cases -- 3.1

In addition to a determination of means to improve stake penetration it was also necessary to determine whether it would be possible for the operator to sustain the effects of increased tool momentum and control the tool.

It was theorized that the velocity of the recoilling mass might be more significant to the operator than the actual impulse. To determine this, a weight was added to the tool which doubled the total tool weight, since momentum had been doubled, recoil velocity would be the same as originally obtained with the engineering model. During test firing the reactions of three operators were observed to determine their opinion concerning the effect of the recoil. In every case the operators reported the recoil to be approximately equal to that of the engineering model. This being the case, it was decided to repeat the same experiment with the total weight reduced by 25 pounds to a total weight of 25 pounds. The recoil was obviously excessive and no further tests at that weight were conducted. This was further correlation of the fact that recoil velocity was the most important factor in operator reaction since recoil momentum was the same in both cases, yet operator reaction was favorable in one case and unfavorable in the other.

A final check of operator reaction was conducted at a total tool weight of 45 pounds. In this case the operator felt the recoil was not excessive but would result in fatigue after a period of firing.

In summary, this phase of the program, while admittedly brief, did provide a conclusive indication that the efficiency of the ballistic hammer could be improved by a factor of approximately 3 within the framework of the requirements for a hand-held man portable system.

VII. CONCLUSIONS

The scope of this contract stated the following objective for the program:

"Establish the feasibility of employing a rechargeable rocket motor or other high speed technique or device for the emplacing and retrieving of Stake GP-112/G in ice, frozen ground, and hard pan and Stake GP-113/G in normal soils. The device resulting from this effort is intended as an accessory tool which will provide a rapid means of emplacing and retrieving ground stakes when time and effort are of the essance or when conventional methods fail. It is not intended to replace present tool equipment supplied with masts used to support antennas and instruments which employ these stakes. An engineering test model suitable to demonstrate feasibility is the goal of this specification."

It is believed that the program conducted to establish this feasibility has demonstrated that the rapid emplacement and retrieve of ground stakes has been conclusively demonstrated. The engineering model fabricated to demonstrate feasibility has been tested in a wide range of soils and soft rock. Successful emplacement and retrieve has been achieved in every instance including several cases where retrieve by conventional techniques could not be achieved. A significant reduction in emplacement and retrieve time as compared to sledge hammer techniques was recorded. Average emplacement operating times of 32 seconds were obtained in normal soils and 35 seconds in stone, average retrieve times of 9 seconds in normal soils and 15 seconds in stone were recorded.

In determining the approach which would best satisfy the program objectives, a broad range of devices were considered and objectively evaluated; these included rocket techniques, conventional and special recoilless techniques, explosive shaped charges, pocket wrenches, vibratory techniques, drills, vibratory drills, and a ballistic hammer.

The results of this investigation indicated that the approach with the greatest promise was the ballistic hammer. The results of tests, first with the breadboard model, and then with the engineering model, substantiated the selection of this approach.

At all times the safety and confidence of the operator were considered. The use of grenade launcher cartridges (blanks) permitted use of an easily controlled energy source which was an item familiar to most Army personnel. This approach also has obvious logistic advantages. The reactions of several operators to tool recoil were carefully noted and the recoil velocity of the tool (15 fps) established on the basis of operator safety and fatigue. It may be conclusively stated, based upon actual tool use, that the fatigue level associated with use of the tool is negligible. It is several orders of magnitude less than that required to achieve emplacement or retrieve by means of the standard 8-pound sledge technique.

The noise level associated with the tool operation was controlled to the extent that it was not unpleasant to the operator and could not be detected by anyone not in the immediate area of the tool's use.

Performance of the 30 caliber carbine M-1 action was adequate to show that a gas operated semiautomatic feed and control system was feasible and should be incorporated in the design. The engineering model employed a standard gun action (M-1 carbine) to achieve rapid semiautomatic firing of the .30 caliber blank cartridges which are utilized as the energy source. An existing action was employed to achieve economies of effort and to maintain the program schedule.

The results of this program indicate that the objectives of rapid emplacement and retrieve can be achieved with a lightweight device. The original design goal was 25 pounds; the emplacement mode came extremely close to this value. The retrieve adaptor ultimately developed added approximately 19 pounds to the tool weight, however, performance was so radically improved over the concepts evaluated earlier in the program that the additional weight was compensated for. The total system weight for the engineering model is within the range of man portable items and can be carried by back pack or in satchel fashion. It should be noted that it appears possible to incorporate the retrieve principle presently employed into a more advanced design to achieve a further weight saving. (The question of tool weight will be discussed further in the Recommendations section of this report.) Performance of this action was adequate to show that a gas operated semiautomatic feed and control system was feasible and should be incorporated in the design.

The engineering model employed a standard gun action (30 caliber carbine M-1) to achieve rapid semiautomatic firing of the 30 caliber blank cartridges which serve as the energy source. An existing action was employed to achieve economies of effort and to maintain the program schedule.

Since the device will utlimately be employed in regions close to enemy forces, it was important that the signature of the device should not disclose its position. Tests to evaluate the visual and aural signature were conducted; results of these tests showed little probability of detecting the tool beyond approximately 200 yards. This was about the same distance that the sledge could be heard.

Extensive tests were conducted to determine that ballistic emplacement had no adverse effect upon stake holding power. Tests were conducted in soils available in Northern Baltimore County and Western Harford County, Maryland, and in every situation measured, the holding power of stakes emplaced with the engineering model was comparable with those obtained when the stakes were driven by sledge. The same results were obtained in local limestone known as Cockeysville Marble.

The Ballistic Hammer is relatively simple from a functional standpoint and requires no specialized operator training or special skills. It can be stated that it actually takes a greater degree of skill to successfully employ the sledge. At no time during the program was there any difficulty in handling or controlling the Ballistic Hammer.

Maintenance of the device is not cumbersome and it appears that the level of maintenance is comparable to that required for a rifle or carbine.

Because the energy source employed is a standard .30 caliber grenade launcher cartridge, there is no problem of logistic support. These cartridges may be drawn from ammunition depots in the same manner as other standard ammunition. It may be shipped and handled in the same manner as small arms ammunition and introduces no unusual hazards.

Finally, it has been proven feasible to emplace and retrieve stakes rapidly and with minimum effort employing the ballistic hammer. The approach is simple, safe, reliable, and produces little signature. This approach, when fully developed, will provide Signal Corps personnel with the capability to significantly reduce the time and effort required to erect or dismantle communications masts, towers, or similar structures.

VIII. RECOMMENDATIONS

The program now concluded has examined the possible approaches to rapid emplacement and retrieve of ground stakes. The superiority of the Ballistic Hammer as a means to achieve the desired goals was established from analytical and experimental data. The feasibility of the ballistic hammer was demonstrated during the test phase of this program.

In summary it is recommended that the following items be considered:

A. Design of an action specifically adapted to the ballistic characteristics of the Ballistic Hammer.

B. A second generation Ballistic Hammer design to fully exploit the many advantages of the approach.

C. An investigation to determine the optimum relation between Ballistic Hammer weight, piston weight, piston velocity and maximum stake driving effectiveness.

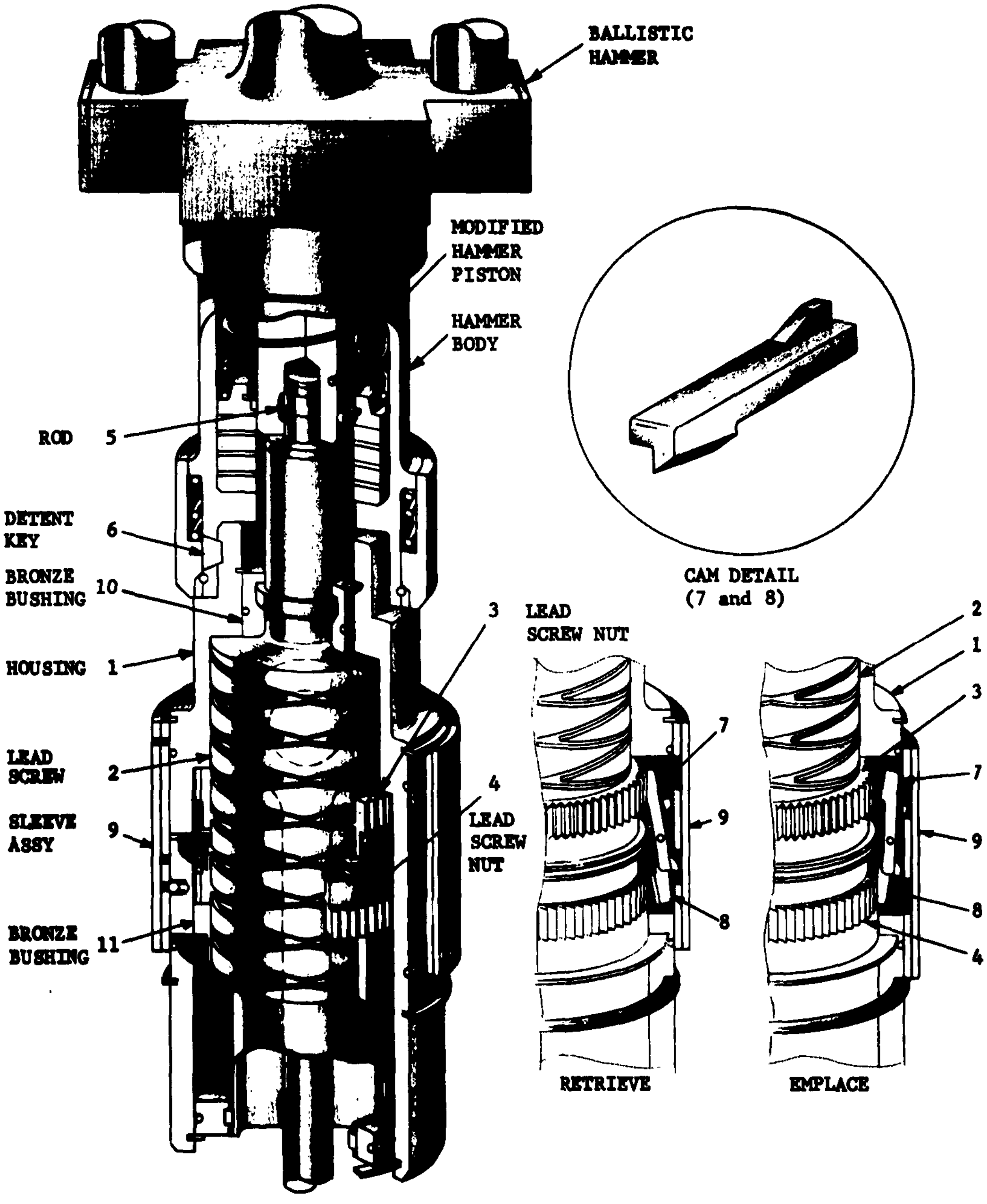
As a result of the work conducted, it is now possible to formulate an approach for an item with significantly improved operating characteristics which could be issued to operating personnel for field use. This device, which might be termed a "second generation", would incorporate the operating principles proven feasible during this program. In addition, it would include increased penetration per blow, further reduced operating time, optimized weight distribution, significant weight reduction, an improved ballistic action design specifically for this tool application and incorporation of production engineering precepts. In addition, it appears that it should be possible to adapt the Ballistic Hammer concept to include the driving and

retrieve of screw type anchors which are widely used as guy anchoring devices. A very limited amount of testing has been performed by AAI to measure torques required to emplace screw anchors. Anchors in the 3 to 5 inch plate diameter range appear to be well within the energy capabilities of the ballistic tool. Figures 52 and 53 show a concept which could be employed with the present engineering model design.

One of the most important pieces of information obtained during the program just completed was the relation of hammer momentum to stake driving effectiveness. Initial testing had been limited in the range of hammer momentum considered because of the total tool weight limit of 25 pounds. Late in the program, however, the indications of the momentum-efficiency relation were so strong that tests were conducted with heavier hammer weights (8#) at the same velocity (116 ft/sec) employed in the engineering model. Results of these tests have been extremely encouraging and show that it is possible to reduce the number of blow required for emplacement by a factor between 3 and 4. It therefore now appears quite feasible to achieve penetration under worst soil conditions in 15 to 20 blows with an average of approximately 9 blows per emplacement in normal soils. Corresponding improvement in the retrieve would also be expected. Tests with the breadboard have demonstrated both the increased driving effectiveness and tolerable recoil (comparable to that obtained with the engineering model). It should be noted that it should be possible to obtain any increased weight required to control recoil momentum by incorporating a hollow jacket around the tool which could be filled with sand, dirt or liquid when the tool was in use and left empty for transportation or storage.

Concept for Hand Held Screw Anchor Adapter
Shown With Present Engineering Model

Figure 52



Screw Anchor Adapter

Figure 53

A fact which strongly influenced the shape and weight of the model was the fact that the semiautomatic cycling action was provided by employing an existing ordnance component (.30 caliber carbine M-1) with minimum modification. This was a very logical choice based upon schedule consideration and the fact that this was a feasibility program. It is fairly obvious, however, that this action leaves something to be desired in any optimum design; it accounts for half the length of the engineering model, is not as rugged as desired, and it was not designed to function based upon the internal ballistic characteristics of the tool.

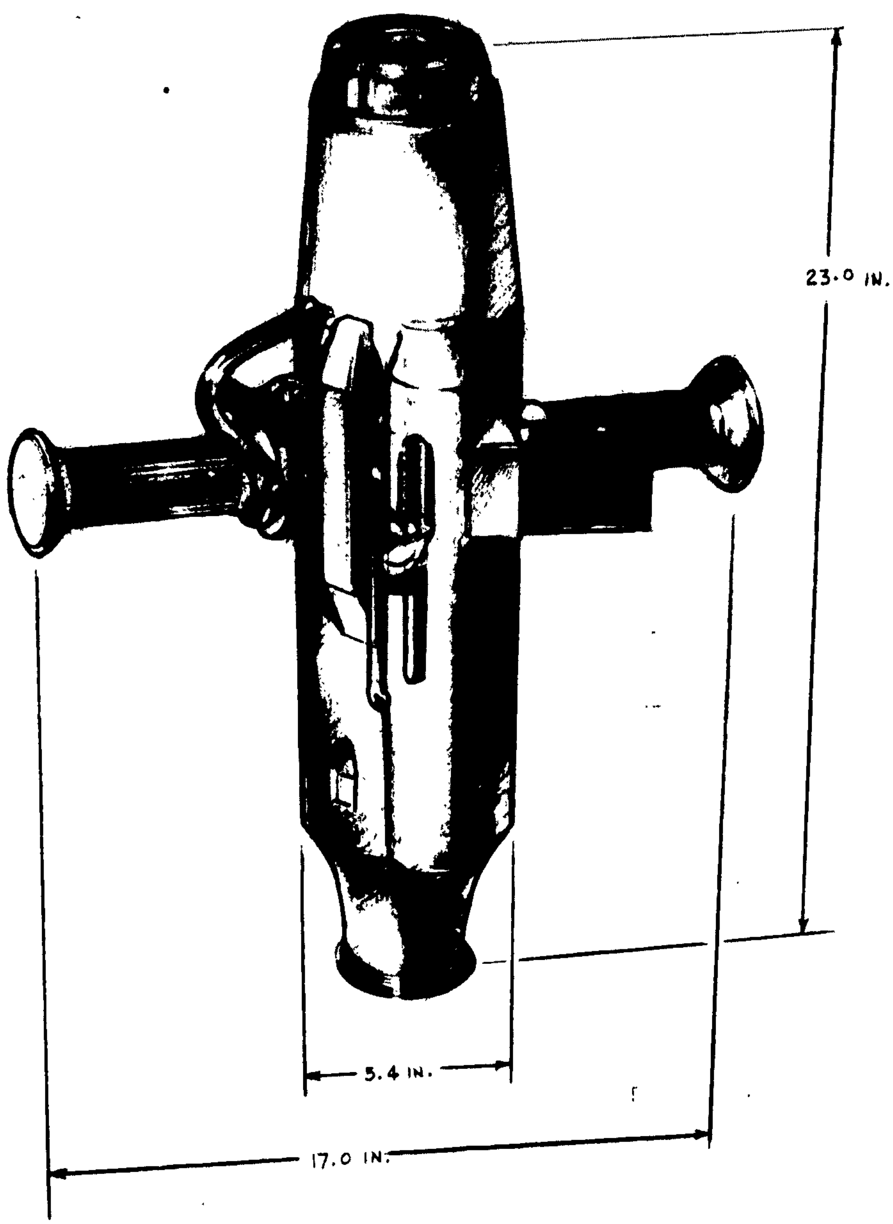
It is recommended that an action be designed for this specific tool application to reduce size and weight and improve performance characteristics. Further, as is shown in Figure 54 , the action might be located on the side of the tool to achieve a size reduction as well as functional advantages. The external configuration of the tool should be optimized from the standpoint of handling ease, protection (elimination of exposed components), accessability for maintenance and manufacturing economy.

Figure 55 illustrates two possible configurations, one with the action on the longitudinal centerline and one on the side of the tool. The sketches shown illustrate a fiberglass tool roll carrier concetp which might offer advantages over the canvas tool roll presently supplied. Some handling difficulty has been noted with a canvas tool roll due to lack of rigidity. In addition, a greater degree of protection would be provided by the plastic carrier. Provision could be made for the item to be carried suitcase fashion or on the back.

By designing the action into the side of the tool, it would be possible to employ a double acting piston which would be capable of acting as a driver in one position and would perform the retrieve operation by simply turning the tool over and, in effect, hammering the stake out. This is the same principle employed in the present retrieve adapter but would perform the function without the weight of a special adapter.

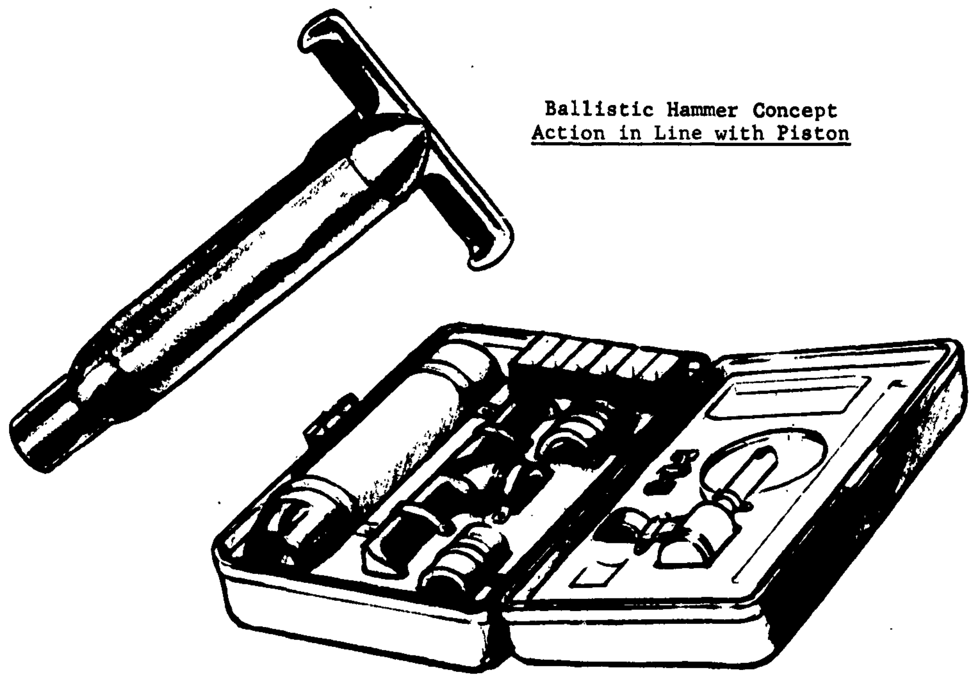
Figure 56 illustrates this concept; figures 57 and 58 illustrate operation.

In summary it would appear desirable to initiate a second generation design effort to fully exploit the many unique features and advantages of a ballistic hammer. Prior to the effort, or in conjunction with it, additional research should be conducted to determine an optimum piston weight, tool weight, piston velocity relationship to achieve maximum stake driving effectiveness.



Second Generation Ballistic Hammer Concept

Figure 54

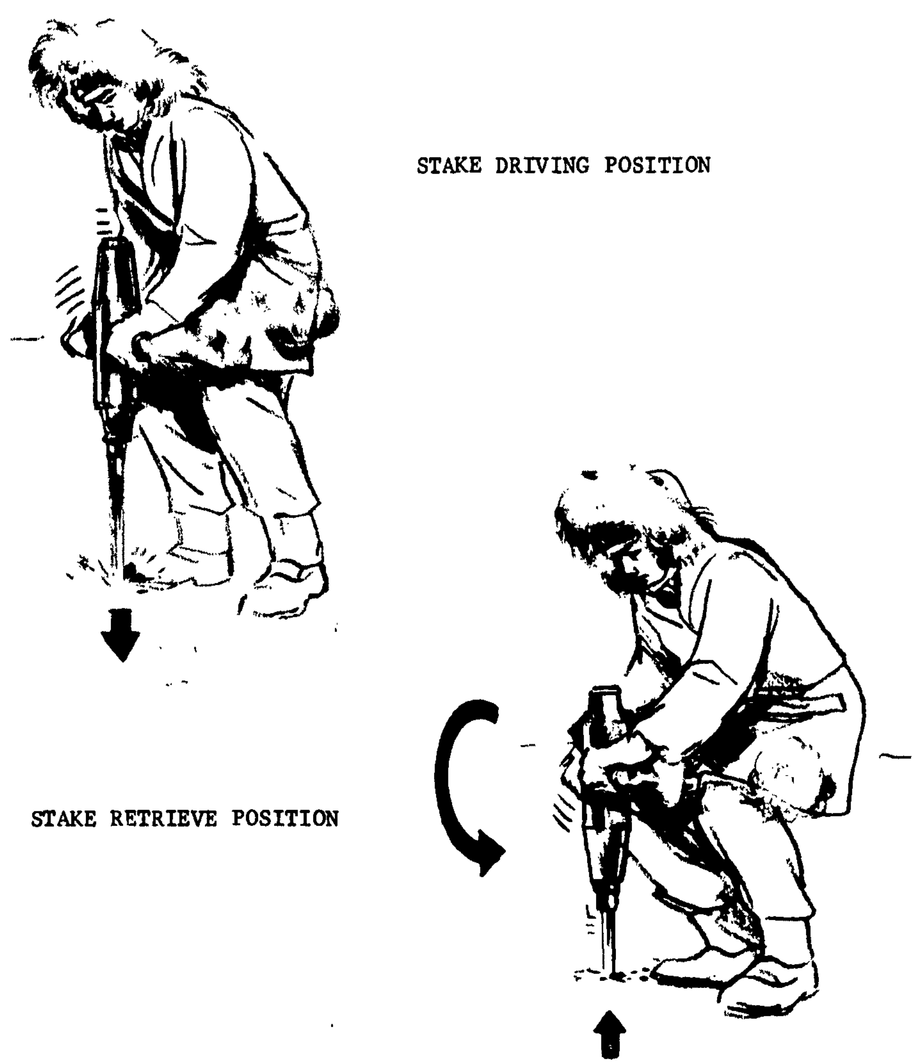


Ballistic Hammer Concept
Action in Line with Piston

Ballistic Hammer Concept
Action Located Parallel to Piston

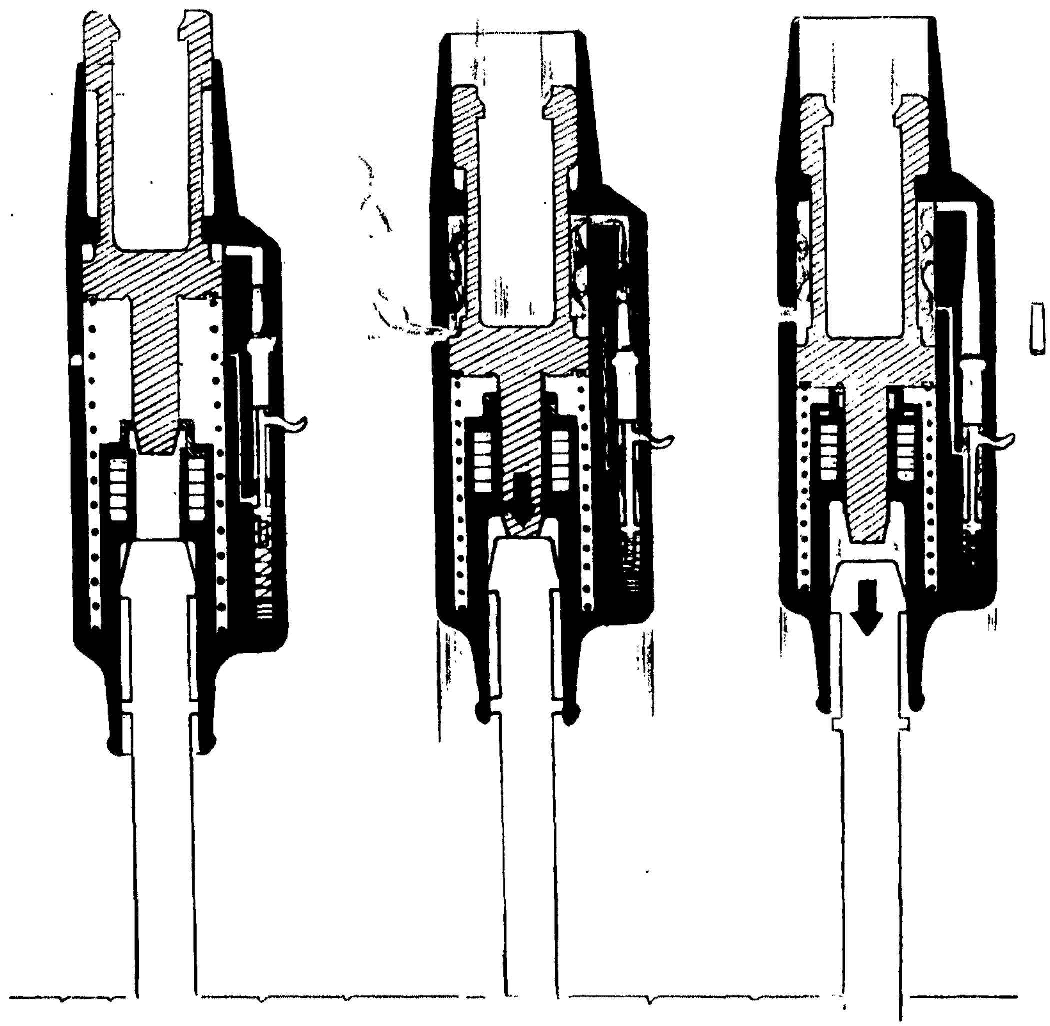
Second Generation Ballistic Hammer Concepts

Figure 55



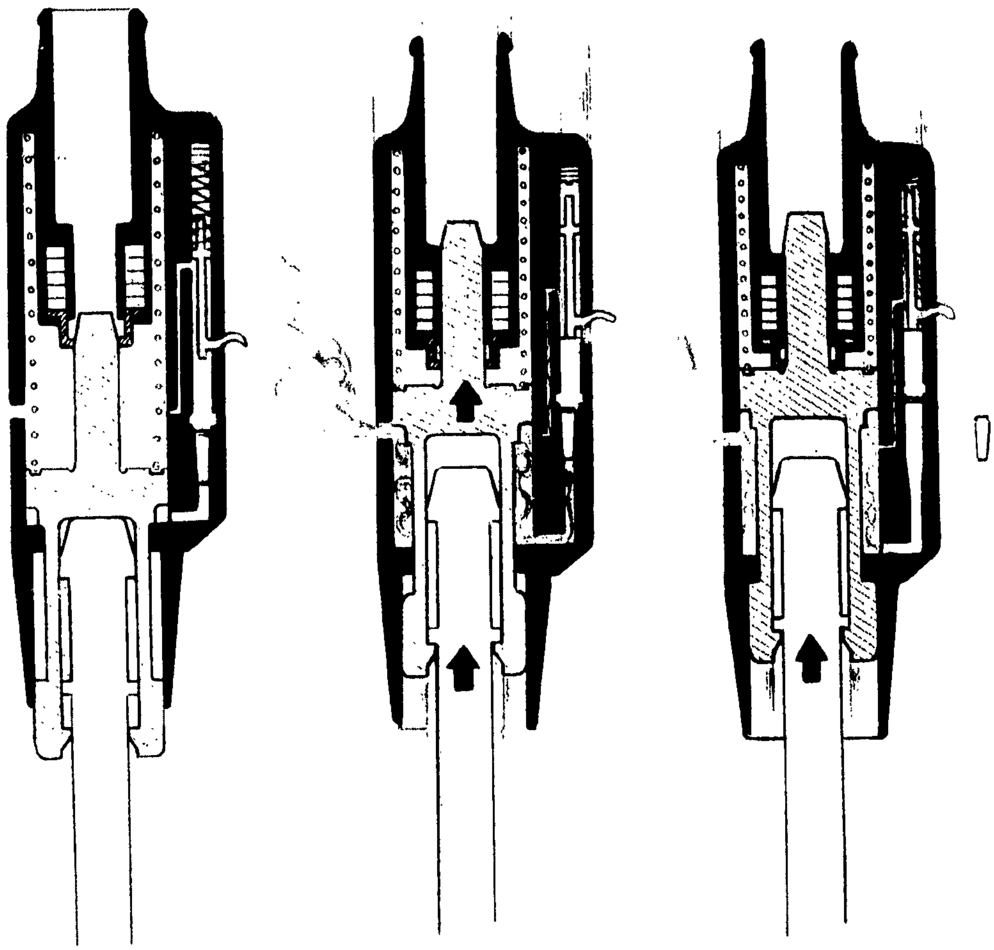
Second Generation Ballistic Hammer Concept

Figure 56



Reversible Concept
Ballistic Hammer in Driving Position

Figure 57



Reversible Concept
Ballistic Hammer in Retrieve Position

Figure 58

IX. IDENTIFICATION OF KEY TECHNICAL PERSONNEL

Brief resumes of key technical personnel who have contributed to this program are shown on the following pages.

Irwin R. Barr, Vice President - Development

Mr. Barr is an original member of AAI, first serving as Chief Design Engineer, and attaining his present position in 1960. Between 1952 and 1960, he directed the company's ordnance design activities in the capacity of Chief Ordnance Engineer. He has had extensive experience in both ordnance and missile development, being co-inventor of the Viking rocket control system and holding patents on a variety of ordnance items, including a vehicle, turrets, bearings, special fin stabilized ammunition, and several automatic guns. Outstanding examples of ordnance material developed under his direction at AAI include the T175E1 Dual Purpose Machine Gun, a series of turret-type cupolas, and the T92 Light Tank.

Mr. Barr is a graduate in Aeronautical Engineering (1940) from the Casey Jones School of Aeronautics. He worked for two periods, 1940-1944 and 1946-1950, for the Glenn L. Martin Company, primarily in missiles. During the last of these two periods he served in the U. S. Army Air Force, receiving the Army commendation medal for design of aircraft rocket launchers.

Nicholas J. LaCosta, Manager, Explosive Ordnance Department

Education: Casey Jones School of Aeronautics, A. E.

Work History: 1959 to present, Aircraft Armaments, Inc.
Manager, Explosive Ordnance Department.
Responsible for directing and coordinating all activities of the Explosive Ordnance Department. These activities include the study, design, development, test and manufacture of systems and components involving propellant actuation, interior and exterior ballistics, terminal ballistics, kinematics, gas dynamics and explosive train design, applications and feasibility studies.

1951-1959, Aircraft Armaments, Inc.
Project Manager responsible for stabilized ammunition development program, Corporal Warhead development, a number of explosively actuated bomb racks, canopy removers, thrusters, and initiators and control systems. Recent responsibilities also include new infantry weapon concepts, recoilless rifle program, special weapons tools and an automatic sequencing device.

1939-1951, The Glenn L. Martin Company
Layout Engineer, Group Engineer, Design Specialist

1938, Bell Aircraft Company. Layout Engineer

1936-1937, Consolidated Aircraft Company. Draftsman

Theodore G. Stastny, Senior Engineer

Education: Georgia Institute of Technology, BME, 1958
Georgia Institute of Technology, MME, 1959

Work History: 1958 to present, Aircraft Armaments, Inc. Responsible for analytical and experimental evaluations of the internal ballistics of recoilless rifles and both hybrid and solid-fuel rockets. Study and design of various cartridge actuated devices employed in ejection mechanisms, parachute deployment, and separation problems. Trajectory, stability and aerodynamic analysis of projectiles and gun boosted rockets. Stress and cycle analysis, including dynamics and kinematics of automatic weapons. Responsible for compiling theories of detonation and explosive phenomena, and fragmentation of bombs and grenades for handbook usage. Designer of "Lo to Hi Pressure Admission Valve" of hot gases (Pat. Pend.). Heat transfer and stress analysis of large, high pressure flat-end chemical reactor. Investigation of underwater ballistics on Mach 0.1 to 0.5 projectiles covering velocity decay, vaporization bubble, stability and nose configurations. Responsible for extensive literature survey and analysis covering techniques of rapid ground anchor emplacement devices including related studies of soil mechanics and soil properties.

1954-1958, Koppers Company, Inc., concurrent with schooling. Design and study of gas cleaning equipment, design of gear teeth for coupling misalignment, studies of sealing ring and piston ring problems.

Technical Society: Member and past officer of American Rocket Society, Maryland Section.

<u>Richard G. Strickland</u>, Principal Development Engineer

Education: City College of New York, B. S., 1953
University of Maryland, Graduate Work

Work History: 1956 to present, Aircraft Armaments, Inc.

Project Manager responsible for design and development of tools and procedures for special weapons handling and disassembly, underwater timing device, and special devices for underwater recovery operations. Design of special tools for remote handling of explosive devices. Survey and study of CAD in all U.S. aircraft and missiles. Experimental study of recoilless rifle internal ballistics; study of recoilless weapon optimization. Spotting round development. Study of missile fuel spark compatibility. Development of timer concepts, applied explosives research. Design and development of underwater guns and ammunition, measurement of effects of underwater explosions. Responsible for development and fabrication of MK25 Demolition Firing Device and associated test equipment. Study and development of ground stake emplacing device and earth anchor concepts.

1954-1956, U. S. Navy Special Weapons Disposal School. Special Weapons Instructor. Nuclear Physics, Health Physics, Weapon Principles, Weapon Configuration and Weapon Effects.

1953-1954, U. S. Navy
Special Weapons Disposal Officer. Field test of special weapons. Attended Nuclear Weapons Training Schools at Field Command AFSWP, Picatinny Arsenal and Lowry Air Force Base.

Arthur C. Powell, Senior Engineer

Education: Massachusetts Institute of Technology, B.S. in M.E., 1948

U.S. Navy Electronics Technician's Schools, 1944-1945.

Work History: 1958 to present, Aircraft Armaments, Inc. Senior Engineer. Design and test of a parachute sequencing device, a hydraulic power supply, recoilless weapons, cartridge actuated devices, a remote arming device, escape system components, hydraulic timers, and production tooling.

1952-1958, Van Zelm Associates, Inc. Design Engineer on a variety of aircraft arresting gears, target drone catapults, and bridle arresting equipment. Field test engineer on aforementioned equipment.

1949-1952, Friez Instrument Division, Bendix Aviation Corporation. Design and test engineering on numerous weather, recording, and electrical instruments.

1948-1949, Industrial Research Laboratories. Design, construction, and test of a filled-bottle goods inspection machine.

Edward T. Kusterer, Chief Structures and Dynamics Engineer - Mechanical

Education: The Johns Hopkins University, B.S. in M.E., M.S. in M.E.

Work History: 1953 to present, Aircraft Armaments, Inc. Responsible for the technical supervision of all analytical work within the Mechanical Engineering Department. The technical fields encompassed in this assignment include Structural Design and Analysis, Dynamics, (in particular the response of structures to transient loadings), Thermodynamics, Aerodynamics and Heat Transfer. Project Manager of a feasibility study of a high "g" centrifuge for the Signal Corps at Ft. Monmouth, New Jersey. The centrifuge was required to impose a 20,000 g's on a 4 pound test item. During this study, investigations were conducted on the structural, dynamics and flutter characteristics of the centrifuge. Consideration was given to operating in a vacuum. Control system, safety devices and drive systems were evaluated and designed. In addition, installation design, and design of the building housing the facility were also furnished. Project Manager for the design and fabrication of components of a nuclear propulsion system for ANPD-G.E. During this program, investigations were carried out in the areas of heat transfer, both transient and steady state, thermal stresses in light weight structures and mechanical design for elevated temperatures. A study of the techniques for fabricating high temperature alloys was also conducted in conjunction with this program. Other assignments included Project Manager for special purpose store for installation on military aircraft, carried out under the auspices of W-PAFB; modification to incorporate chemical filled munitions into the CORPORAL warhead. Project Manager

Edward T. Kusterer, Chief Structures and Dynamics Engineer - Mech. (cont.)

in charge of the design of a spray tank to be carried externally on aircraft of the A4D or FJ4B type, design and analysis of armored vehicles, including a 2,000 pound wheeled vehicle and the T-92 light tank. Responsible for packaging and installation design of electronic equipment.

1949-1953, The Johns Hopkins University, Department of Radiology. Design of high speed x-ray camera for installation in a centrifuge at the Naval Air Development Center, Johnsville, Pennsylvania.

Publications: AAI Report, "Experimental Determination of the Stability of Conical Shells," (submitted for publication in the Journal of Applied Mechanics).

AAI Report, "Feasibility Study of a High G Centrifuge - Final Report.

Thesis: "Orthogonally Stiffened Circular Plate with Clamped Edges Subjected to Uniform Lateral Pressure."

Charles H. Gonnerman, Engineer

Education: Drexel Institute of Technology, B.S. in M.E., 1962

Work History: 1962 to present, AAI, Assistant Engineer.
Layout and detail design, stress analysis, and test work concerned with the Chemical Corps' Chemical Reactor, the Signal Corps' Stake Emplacement and Retrieving Device, and their related components. Designed two machines which enabled the study of motion effects on the performance of the Chemical Reactor. Layout design, stress analysis, and weight analysis of projectiles and various weapon components for proposal usage.

1958-1962: Chemical Research and Development Laboratories Army Chemical Center; concurrent with schooling.
Study, design, and test of explosive disseminating devices and related components; trajectory and aerodynamic analysis of projectiles.

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Cockeysville, Maryland

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by R. G. Strickland and C. Gonnerman
Final Report, June 1962 to November 1963
230 pages, 75 figures, 16 references

Unclassified Report

The Final Report discusses the selection of the design approach, preliminary testing, detail design, stress and kinematic analysis, and test programs conducted with the engineering model of the ballistic hammer for emplacement and retrieve of ground stakes. The chronological history of the program is reviewed and conclusions and recommendations are presented. A supplement to the report has also been included which serves as in instruction manual for the engineering model.

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Cockeysville, Maryland

RAPID EMPLACING & RETRIEVING DEVICE FOR GROUND STAKES GP-112/G AND GP-113/G
by R. G. Strickland and C. Gonnerman
Final Report, June 1962 to November 1963
230 pages, 75 figures, 16 references

Unclassified Report

The Final Report discusses the selection of the design approach, preliminary testing, detail design, stress and kinematic analysis, and test programs conducted with the engineering model of the ballistic hammer for emplacement and retrieve of ground stakes. The chronological history of the program is reviewed and conclusions and recommendations are presented. A supplement to the report has also been included which serves as in instruction manual for the engineering model.

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